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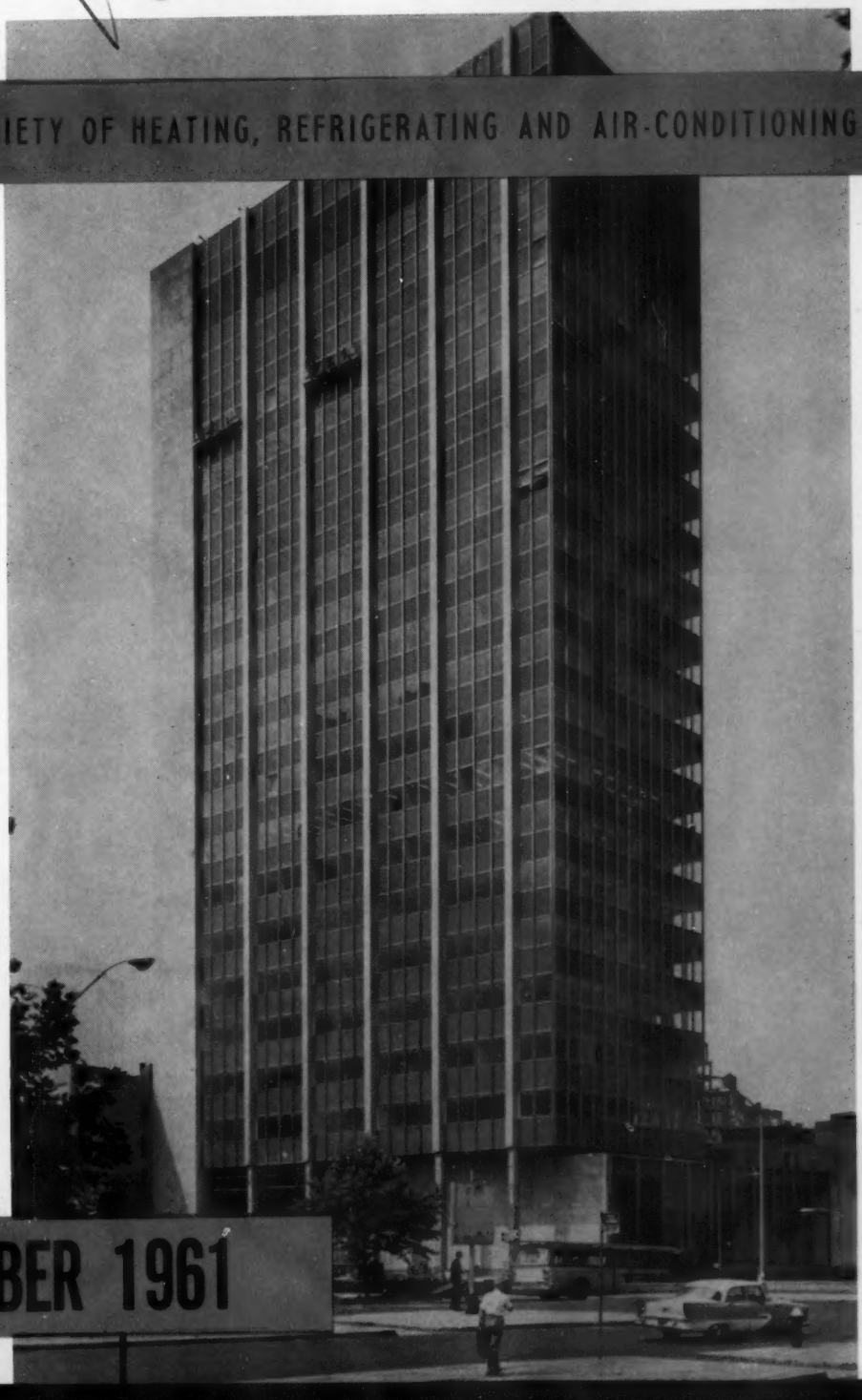
JOURNAL

Heating • Refrigerating • Air Conditioning • Ventilating

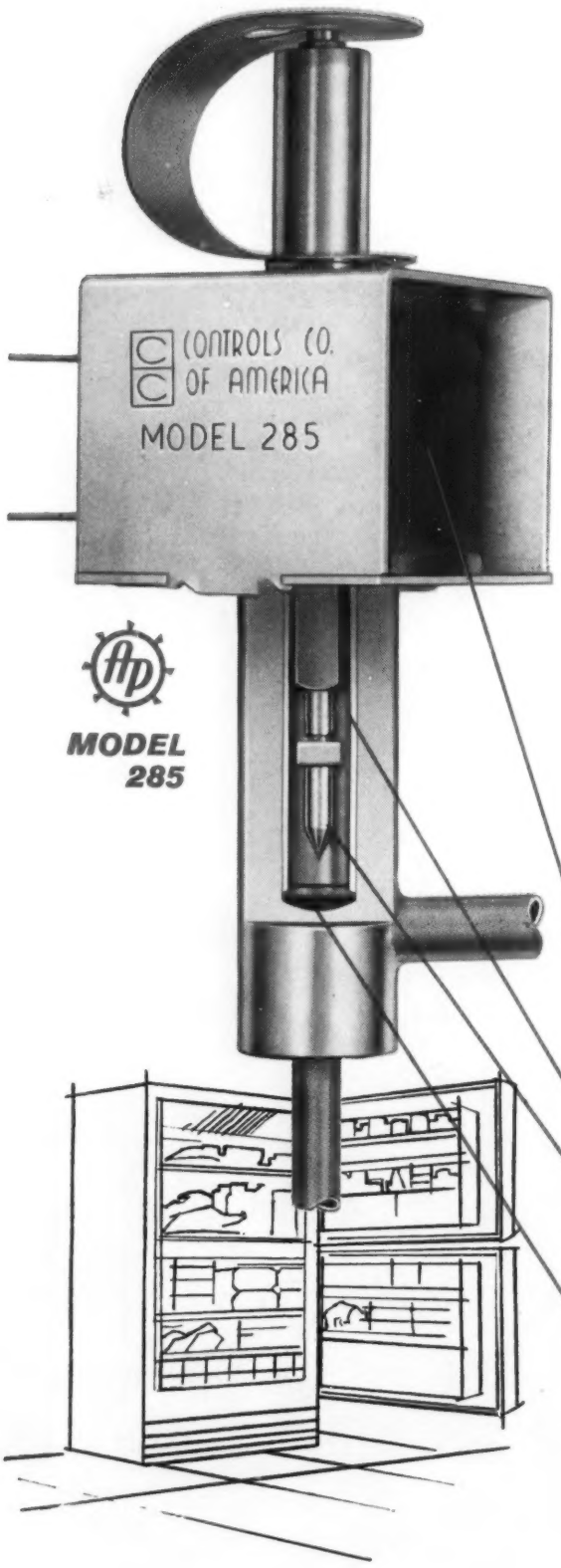
AMERICAN SOCIETY OF HEATING, REFRIGERATING AND AIR-CONDITIONING ENGINEERS



UEC
NEW HOME
OF
ASHRAE

SEPTEMBER 1961

Open and shut case for the only valve that offers "nonstick" dependability for refrigerators utilizing hot gas defrosting



This new Model 285 hot gas defrost valve always opens and closes on schedule, making automatic defrosting of domestic refrigerators and home freezers absolutely foolproof. Everything possible has been done by Controls Company's engineers (see cutaway) to prevent "hanging-up" and the food spoilage that results from such valve defection. That's why if you're considering hot gas defrosting, your opening move is to investigate the new A-P Model 285. Write today for full facts.

Here's how A-P engineers solved a sticky problem

SOLENOID lifts needle . . . gravity plus pressure drop closes it. Solenoid coil is encapsulated with a layer of EPOXY resin — moisture resistant for long life. Coils available with open yoke (shown) or total metal enclosure.

PLUNGER TUBE is stainless steel with "mirror-smooth" internal surface finish. Highest corrosion resistance.

STAINLESS STEEL NEEDLE slides smoothly without sticking. Head of needle is radiused to give one-point contact in open position. Edges of plunger are rounded for minimum friction and long life.

VALVE INTERIOR is polished to within 20 rms for minimum coefficient of friction. Seat as well as needle is of stainless steel adding to no-stick, long-life operation.

MISCELLANEOUS SPECIFICATIONS: 3/32" orifice. 225 lb. lift. 200 cc/minute max. leak at 200 lb. dry air. Coils for 115- or 230-volt, 60-cycle service.

Creative controls for industry

CONTROLS COMPANY OF AMERICA

HEATING AND AIR CONDITIONING DIVISION

2456 N. 32nd Street, Milwaukee 10, Wisconsin • Cooksville, Ontario • Zug, Switzerland

HAC-18-60

SEPTEMBER 1961

VOL. 3 NO. 9

OFFICIAL PUBLICATION

ASHRAE JOURNAL

Formerly Refrigerating Engineering including Air Conditioning, and incorporating the ASHAE Journal.

On the Cover: Late July view of ASHRAE's new UEC Headquarters. See also page 55.

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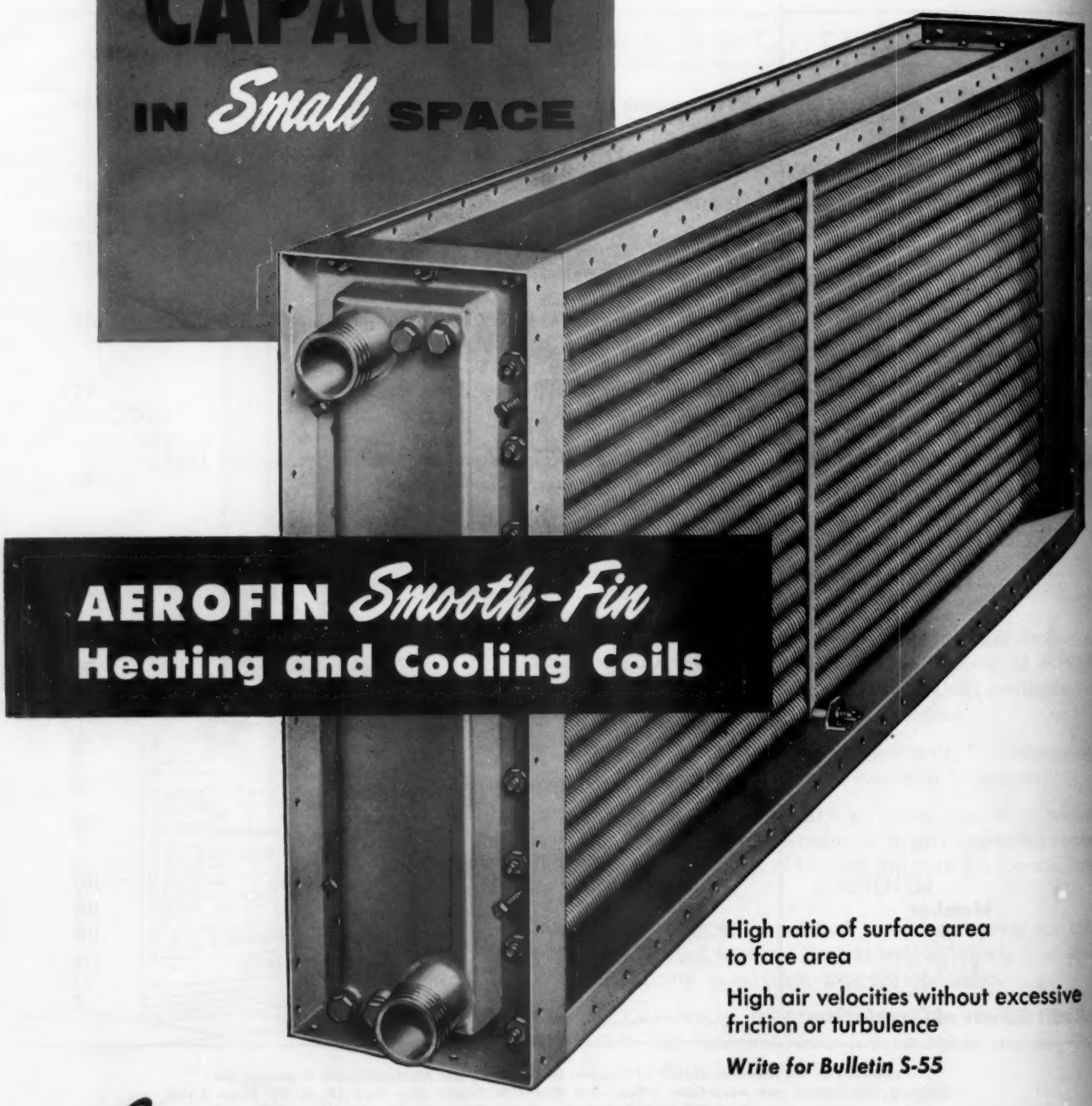
BIG CAPACITY IN *Small* SPACE



AEROFIN *Smooth-Fin* Heating and Cooling Coils

High ratio of surface area to face area

High air velocities without excessive friction or turbulence

Write for Bulletin S-55

AEROFIN CORPORATION

101 Greenway Ave., Syracuse 3, N.Y.

Aerofin is sold only by manufacturers of fan system apparatus. List on request.

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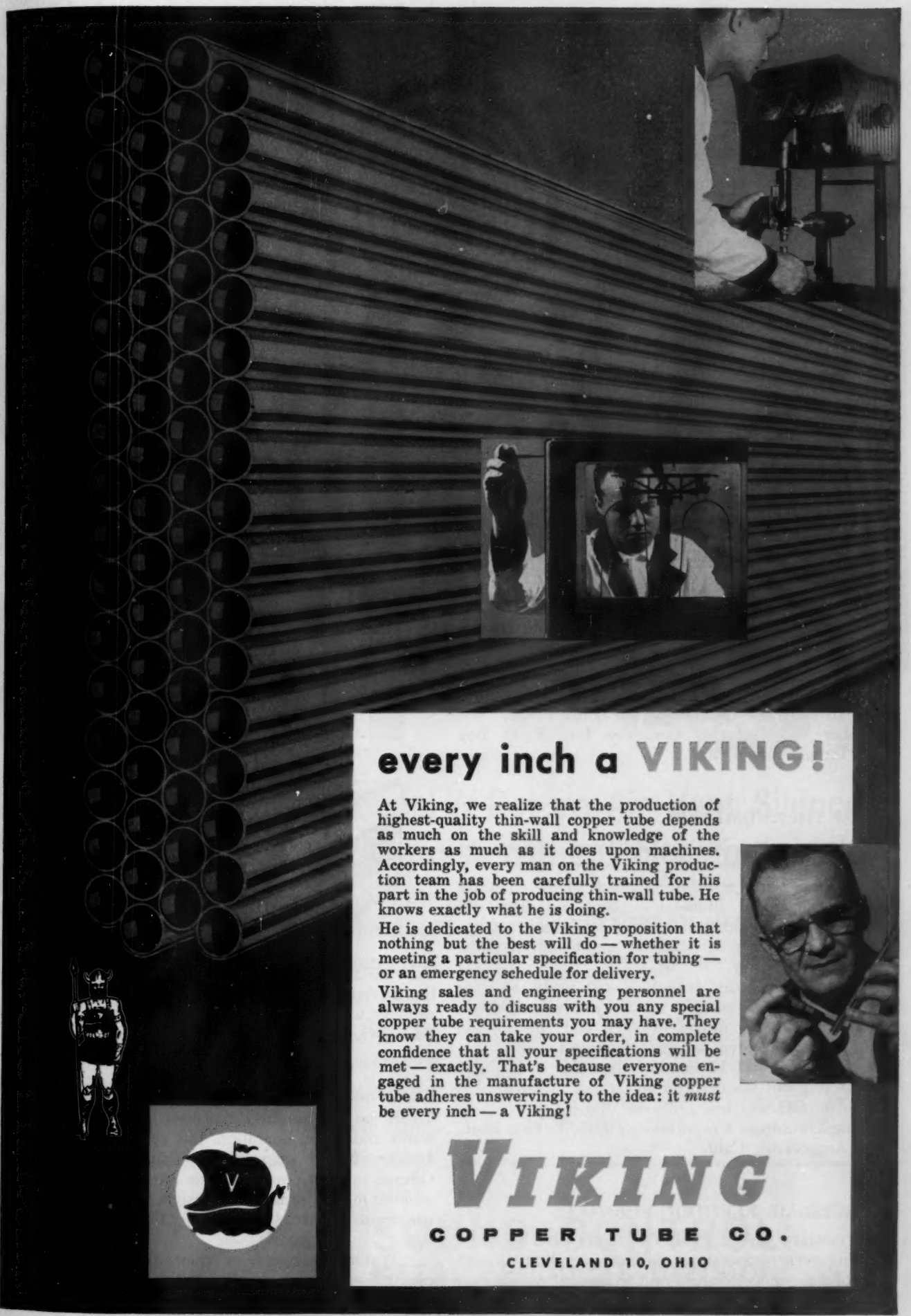


every inch a VIKING!

At Viking, we realize that the production of highest-quality thin-wall copper tube depends as much on the skill and knowledge of the workers as much as it does upon machines. Accordingly, every man on the Viking production team has been carefully trained for his part in the job of producing thin-wall tube. He knows exactly what he is doing.

He is dedicated to the Viking proposition that nothing but the best will do — whether it is meeting a particular specification for tubing — or an emergency schedule for delivery.

Viking sales and engineering personnel are always ready to discuss with you any special copper tube requirements you may have. They know they can take your order, in complete confidence that all your specifications will be met — exactly. That's because everyone engaged in the manufacture of Viking copper tube adheres unswervingly to the idea: it *must* be every inch — a Viking!

VIKING

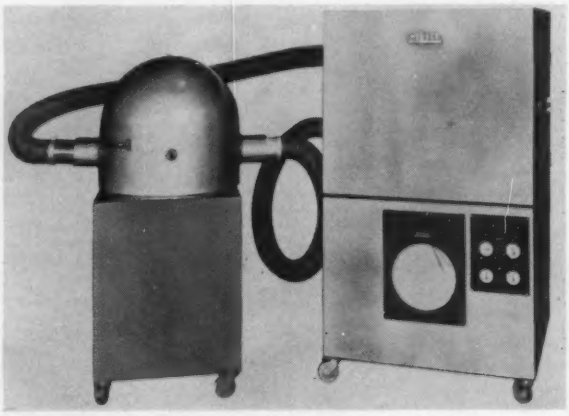
COPPER TUBE CO.

CLEVELAND 10, OHIO

PARTS AND PRODUCTS

PORTABLE CHAMBER

Designed for use on components, parts and products undergoing vibration or shaker tests, and for other applications where it is impractical to bring the item to the chamber, a portable environmental chamber, Model W-2—100+200, has been introduced. A 19-in. diam insulated dome, connected to the mobile chamber with two flexible, insulated hoses, fits over the product. Temperature range is from —100 to 200 F,



with a pull-down from max to min temperatures in 45 min. Internal air circulation is cited as assuring zero stratification throughout the entire range.
Webber Manufacturing Company, Inc., P. O. Box 217, Indianapolis 6, Ind.

STEM THERMOMETERS

Featured on this line are anti-parallax dials, the dial and the pointer being in the same plane. For high pressure use, the base and butt of the stem are inert gas welded. Bi-metal element and drive rod are silicone damped to minimize vacillating readings where vibration is encountered and to afford maximum heat transfer.

Four ranges are available: —40 to 160 F, 0 to 220 F, 50 to 300 F and 150 to 750 F. Over-range protection is 50% for ranges to 300 F and 10% for the range to 750 F. Stem lengths available for all ranges are 2½, 6, 9 and 12 in., including the ½-in. NPT connecting nut standard on all models. Dial head is three in. OD and has a 2⅝-in. dial.
Pacific Transducer Corporation, 11836 W. Pico Blvd., Los Angeles 64, Calif.

FORCED-AIR ELECTRIC FURNACE

Thermostat-controlled, this furnace is cited as maintaining even temperatures without recycling through use of a sequencer control unit. This unit, which may control up to seven heating elements depending on the furnace rating, energizes only enough elements to supply the heat needed to satisfy the thermostat setting. Heating elements are energized at random, one at a time, until the correct temperature is achieved. Two cabinet sizes and six heat ratings, from 34,000 to 119,000 Btu/hr, are offered. Btu ratings vary with the number of heating elements used in the unit. Completely enclosed and insulated with one-in. vinyl-coated glass wool, the furnace can be operated vertically, horizontally or counterflow with but a minor change in the placement of the controls.
Westinghouse Electric Corporation, Air Conditioning Div, P. O. Box 510, Staunton, Va.

UTILITY FREEZER

For backroom storage of frozen foods at —10 F and below, the UF-72 has a 72-cu ft capacity and is a step-in model, providing ready access to the interior. Built and shipped in sections, the freezer has a ¾-hp, sealed plug-in panel refrigeration unit. Compact and self-contained, the compressor can be mounted on the ceiling, so as not to obstruct storage space. Insulation is five-in. glass fiber.
Nor-Lake, Inc., Hudson, Wisc.

SINGLE-INLET MIXING BOX

For use in interior zones requiring cooling the year round, a single-inlet mixing box has been introduced. Temperature control in the area is accomplished by providing more or less cooling, with a constant volume of air delivered at all times. On a thermostat demand for less cooling, the amount of cool primary air is reduced, and secondary air is induced into the pri-

mary air stream from a ceiling return plenum. With the new unit, it is possible to reduce the amount of primary air, proportionately, as much as 50%.
Barber-Colman Company, Rockford, Ill.

RESIDENTIAL STEEL BOILERS

To its gas and oil-fired, cast-iron heating boilers for water and steam operation, this company has added a new line of packaged residential steel boilers. Offered in the Commando line are four gas and four oil-fired models of fire-tube design. Graduated hourly gas inputs are 100,000, 125,000, 150,000 and 175,000 Btu.

Water-type boilers of square design have forced circulation and are supplied with or without tankless heaters to supply domestic hot water. Included are

IMPORTANT ANNOUNCEMENT

to people who hear voices*

*voices that intrude or disrupt and noises that project when they should be hushed.

The new Aircoustat® Model W Return Air-Vent Silencers stop the transmission of noise without blocking air flow

Aircoustat Return Air-Vent Silencers eliminate the distracting sound of voices that spill from one area to another. Their slim design gives you a choice of installation. You can install them within a wall or ceiling or hang them on doors or walls. Let Koppers long experience in sound control help you. Write today for information to: KOPPERS COMPANY, INC., 3509 Scott Street, Baltimore 3, Maryland.

Model W Return Air-Vent Silencer

In sizes for all applications:

Model	Thickness	Width	Length
W-1	3½″	30″	48″
W-2	3½″	42″	48″
W-3	5″	30″	48″
W-4	5″	42″	48″
W-5	7″	30″	48″
W-6	7″	42″	48″

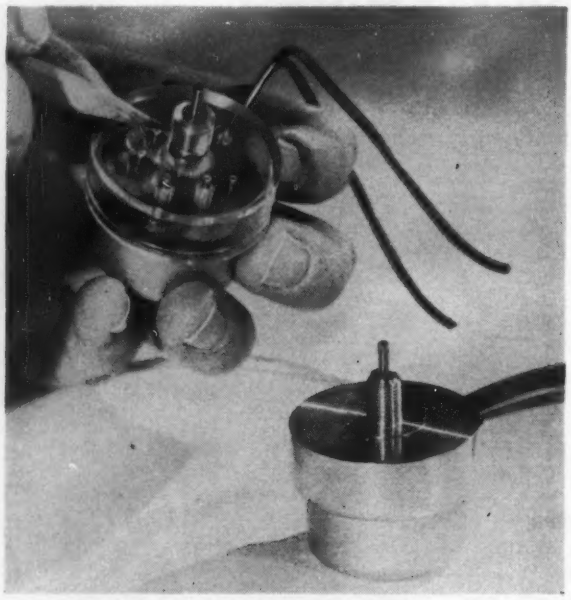
KOPPERS®

SOUND CONTROL
METAL PRODUCTS DIVISION
Engineered* Products Sold with *Service

pump, burner assembly, operating controls, relay and relief valve.
Rheem Manufacturing Company, 7600 S. Kedzie Ave., Chicago 52, Ill.

SYNCHRONOUS TIMER MOTOR

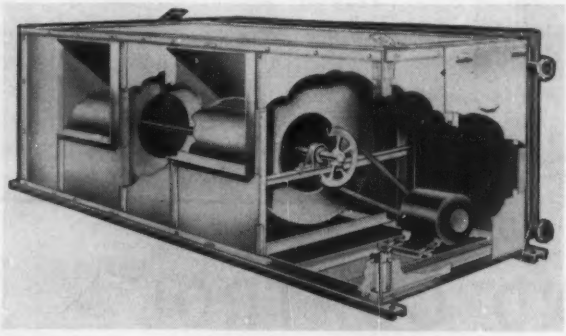
Sealing is effected on this oil-filled timer motor by a specially-designed spring-loaded packing gland, which prevents the oil from working out of the housing when



the motor is operating. Key element of the packing gland is a neoprene O-ring which is held under constant pressure by a spring. Compact, the motor's outside diameter measures but 1-11/16 in. In addition to minimizing wear on working parts, the special mineral oil lubricant in the motor housing assures silent operation. Other key components, also submerged in oil, include a variable pole arm and a magnetic directional device cited as maintaining proper rotation of the gear when the motor is energized.
Lake City, Inc., a subsidiary of Controls Company of America, 110 W. Woodstock, Crystal Lake, Ill.

HORIZONTAL, VERTICAL AIR MOVERS

Expanding its coverage of the commercial and industrial air conditioning fields, this company has



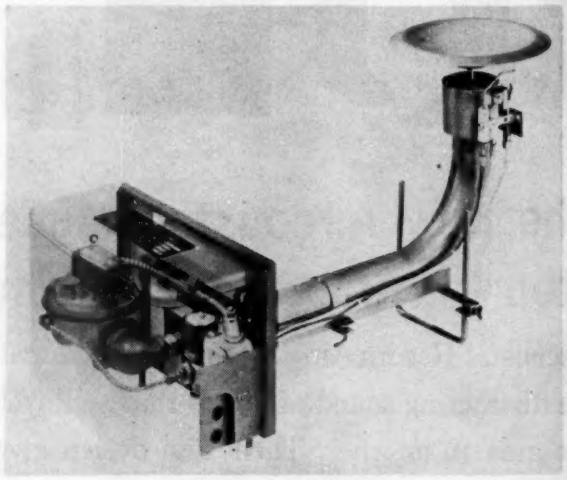
introduced a new line of one and two-fan, horizontal-vertical air movers in the 500 to 35,000-cfm range. Motors are positioned inside the unit; a large, full-entry access door has been provided. By a simple adaption in the field, coils and filter frame may be added externally to the units. Modular construction affords on-the-job flexibility, in that fan sections may be field rotated. Fans are forward-curved, with cast-iron center hub. On motors up to 7½ hp, a variable pitch motor drive is provided. An adjustable motor-mount base and standard belt duty, 1750-rpm, drip-proof motors are part of the basic equipment.
Hi-Press Air Conditioning of America, Inc., Drayer-Hanson Div, 3301 Medford St., Los Angeles 63, Calif.

NEGATIVE ION GENERATOR

To relieve distress of air-borne allergies, Ionitron 5000, a portable air ionizer, generates an abundance of negative ions. Effective in an average size room, it is cited as giving relief at a distance of six to eight ft from the user. Plugging into standard ac lines, it measures 7¾ in. high, 12-5/16 in. wide and 7 in. deep.
Philco Corporation, Tioga and C Sts., Philadelphia 34, Pennsylvania.

GAS BURNER LINE

Additions to the Spread Heat line of plate-type, upshot gas burners extend the line to five models offering a delivery range from 50,000 to 1,000,000 Btu/hr. Three different types of controls will be



available: standing pilot, with 24-volt automatic gas valve and transformers, runner lighter ignition; constant electronic, with 115-volt automatic gas valve, runner lighter ignition, electronic relay and constant pilot; and intermittent electronic, with 115-volt automatic gas valve, automatic spark ignition, 6000-volt transformers, electronic relay and intermittent pilot.
Barber Manufacturing Company, 1052 E. 134th St., Cleveland, Ohio.

SPRAY NOZZLES

Design and inherent low friction of these butyrate injection, molded hollow cone spray nozzles allows uniform spraying to begin at pressures below two

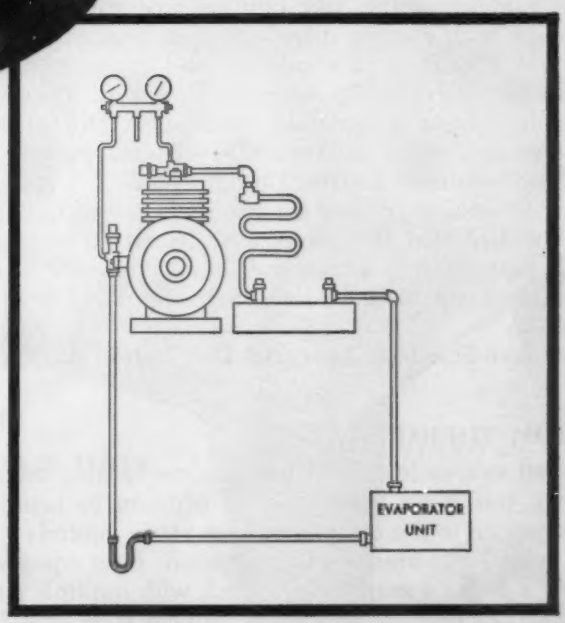
NEW! FROM MUELLER BRASS CO.

Streamline® ONE-PIECE SUCTION-LINE P-TRAPS

FOR EFFICIENT OIL MIGRATION IN REFRIGERATION AND AIR-CONDITIONING SYSTEMS*...

Streamline wrot-copper suction-line P-Traps, just introduced by Mueller Brass Co., answer a definite demand for an effective method of promoting efficient oil migration in modern refrigeration systems. Today, for example, in many large food markets and other outlets employing refrigeration, compressors and condensers are installed on balconies. Such remote units are likely to have long horizontal suction lines or vertical risers exceeding 3' in height which create the problem of unsatisfactory oil return to the compressor. The installation of a Streamline Suction-Line P-Trap quickly and effectively eliminates this problem because the P-Trap drains the oil from the horizontal runs approaching the risers; the oil then migrates back to the compressor, either as a rippling oil film, a mist, or a transparent colloidal dispersion. Vapor velocity can fall as low as 160 feet per minute and satisfactory oil return can still be achieved when a Streamline P-Trap is installed.

* The need for, and the effect of P-Traps on systems having long horizontal suction lines or vertical risers, is fully discussed in Section 66 of the R.S.E.S. Service Manual.



ONE PIECE DESIGN SAVES LABOR AND MATERIALS

The new Streamline P-Trap replaces the cumbersome assemblies of individual fittings with one piece of seamless copper tube, thereby reducing the high costs of labor and materials. The P-Trap is manufactured in 3 popular O.D. sizes; ⅞", 1⅛" and 1⅜", and is ready to be soldered into the line. Use this handy P-Trap in your next installation. Your wholesaler has them in stock. For further information, send for Product Data Sheet No. 14.

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MUELLER BRASS CO. PORT HURON 15, MICHIGAN

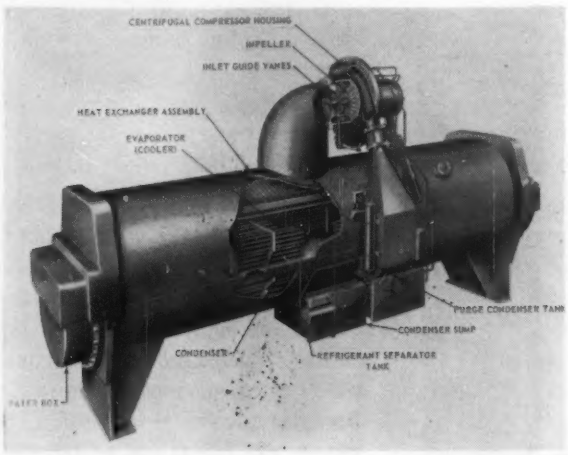
VAMPCO ALUMINUM PRODUCTS, LTD., STRATHROY, ONTARIO • Exclusive Canadian Representative for Mueller Brass Co. • Air Conditioning and Refrigeration Products

psi. Nonporous plastics material is cited as resisting mineral build-up and withstanding temperatures above boiling and below freezing. Nozzles are installed easily by hand. Seven capacities from 0.8 to 7 gpm and pipe sizes of ⅜, ½ and ¾ in. are available for use in cooling towers, air washers, evaporative condensers and similar installations.

Austin Manufacturing Corporation, 1201 W. 24th St., Austin 5, Texas.

COMPACT REFRIGERATION UNIT

Intended for use as the heart of central station air conditioning systems, providing chilled water for operation of zone or individual room conditioners,



this packaged, hermetic centrifugal type Tonrac unit features substantial reductions in size and weight. Available in eight size increments from 50 to 100-ton nominal capacity, the packaged design is completely piped and factory-insulated, requires no auxiliary water piping, refrigerant piping or control tubing installation.

Comprising the basic unit are a centrifugal compressor with electric drive motor, a heat exchanger section consisting of condenser and cooler portions with necessary piping, valves and controls. Power is supplied from a separate motor-generator-exciter power unit which converts 60-cycle line current to 300-cycle current for the Tonrac motor.

An advantage cited for size reduction of the unit is the fact that the completely assembled machine will pass through a standard 36-in. doorway. Total packaged weight of the unit, including power unit, is 6700 lb.

American-Standard, Industrial Div, Detroit 32, Mich.

ROOM THERMOSTAT

Added to this line of hydronic home heating equipment, this room thermostat, in addition to being a companion to the company's zone valve, controls gas, oil and stoker-fired heating systems. It is equipped with a positive snap-action switch with multiple contacts, a low mass, bi-metal actuator for sensitive control and an adjustable heater for primary control systems. Featured are a single-pole, single-throw switching arrangement and low operating voltage, from 20 to 30 volt ac. Unit is adjustable from 0.3 to 0.9 amp. The thermal range is 50 to 90 F, with a differential of 1.5 F.

Taco Heaters, Inc., 1160 Cranston St., Cranston 9, Rhode Island.

PRE-JACKETED INSULATION

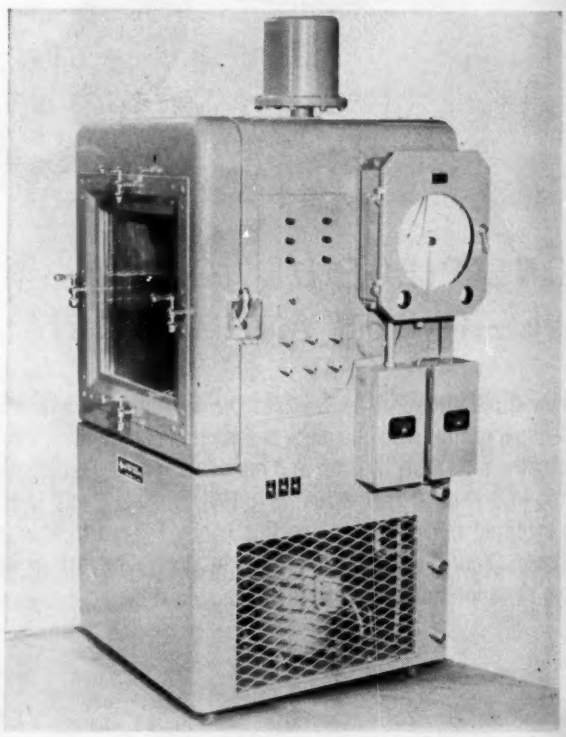
Allowing job-site fabrication of aluminum-covered elbows, turns, sweeps and bends directly from straight lengths, this insulation technique utilizes a preformed band, designated Miter-Seal. These units, available for use with pipe sizes from ¾ to 24 in., allow application of jacketed insulation in many cases where it was not possible except on straight runs.

Fabrication of insulation for a fitting is cited as requiring but a few steps, using a straight length of Metal-On, a Miter-Seal band, banding wrench and a saw. Metal-On is a pre-jacketed insulation system that consists of Thermobestos, an isolation barrier, and a weatherproof aluminum covering, prefabricated for application in one step, by snapping the sections over the pipe and securing with an aluminum snap-strap. Insulation is available from one to three in. thick.

Johns-Manville, Insulation Div, 22 E. 40th St., New York 16, N. Y.

VACUUM/PRESSURE CHAMBER

Designed for testing and calibrating pressure sensing and indicating instruments, Model ASU-40-3-HC provides a pressure range of 20 to 32 in. mercury absolute, over a temperature variance from —40 to 200 F. Containing a three-cu ft working area, the stainless steel



chamber measures 20 in. high, 20 in. wide and 12 in. deep. Six ¼-in. stainless steel connectors and six

McQUAY heavy duty, air cooled Aircon condensers are constructed to give complete, dependable operation for either refrigeration or air conditioning —or both. They are available in horizontal or vertical air flow models in 19 sizes from 7½ to 100 tons nominal capacity in a single unit. Unlimited capacity is available in multiple unit installations. The low profile, quiet operation, flexibility and the economy in installation and operation make the McQuay Aircon ideal for every installation. The McQuay Seasontrol, available as an accessory, offers automatic head pressure control for use any time of the year, winter or summer. For complete information call your local McQuay representative, or write McQuay, Inc., 1606 Broadway Street N. E., Minneapolis 13, Minnesota.

McQuay Means Quality

for AIR CONDITIONING, REFRIGERATION OR BOTH

19 SIZES • 7½ TO 100 TONS IN A SINGLE UNIT

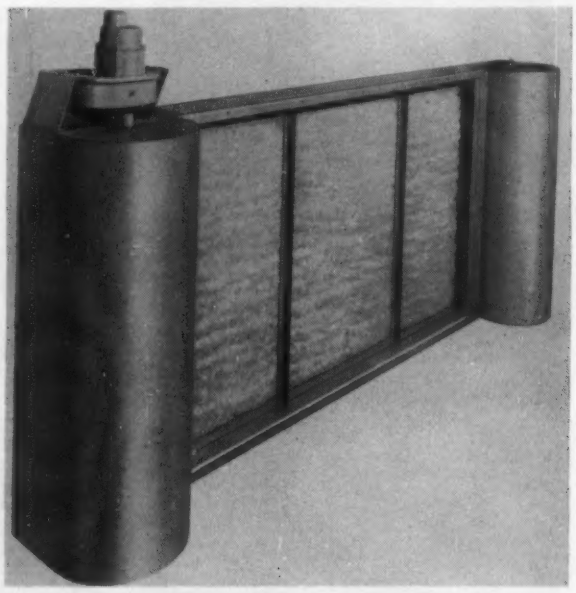
McQuay INC.

AIR CONDITIONING • HEATING • REFRIGERATION

Sealtron, 30-amp terminals are provided with the cabinet.
Cincinnati Sub Zero Products, 3932 Reading Rd., Cincinnati 29, Ohio.

AIR CLEANER

Roll-Kleen Model H-5, a versatile roll-type air filter is available with automatic (motorized and controlled operation) or manual drives, with or without media covers on both ends and is manufactured in 84



standard sizes. As shown in the accompanying illustration, the unit is designed for installation in a variety of positions, regardless of the direction of air flow. Media movement can be in either direction. Frequently used where headroom or vertical clearance is at a minimum, this is installed by direct attachment in the duct.
Farr Company, P. O. Box 90187, Airport Station, Los Angeles 45, Calif.

REFRIGERATION THERMOSTAT

Combining lower internal operating forces with maximum utilization of space to provide smaller size, quieter operation and lower knob torque, the Bulletin 9530 straight range and constant cut-in thermostat has an extensive array of adjustment and differential settings, end or side screw and end or side quick-connection terminations.

Replacing the range calibration spring is a system consisting of a molded drive lever, a contact reed and a flipper spring. These elements are cited as forming a fast break toggle switch mechanism which will not open accidentally even when the contacts are at or near zero force condition.
Cutler-Hammer, 315 N. 12th St., Milwaukee 1, Wisc.

PORTABLE ELECTRIC HEATERS

Featured on all three new models offered are safety tip-over switch, Fast-Glo heating ribbon elements, tamper-proof safety grille, six-ft heater cord, hooded feet for protecting floors, large carrying handle and thermostat control.

Model 107 offers two heat ranges: 1200 and 1500 watt. Thermostat control permits dialed temperature within each range. A two-position switch enables selection of either range and permits quick heat pick-up on high and lower sustaining heat for long periods on low.

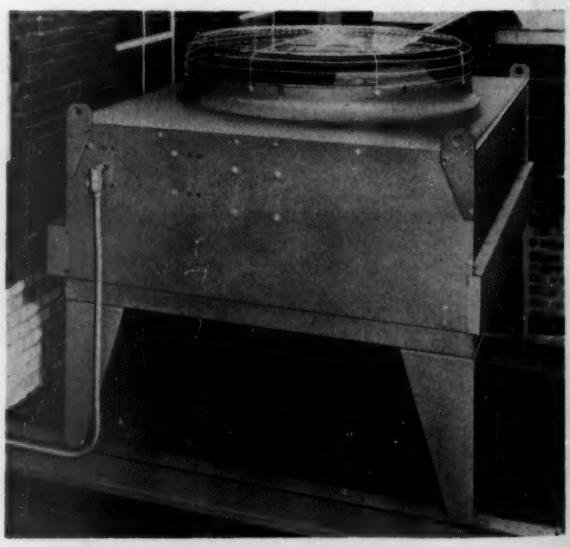
Suited for supplementary heat in a home, Model 106 has a capacity of 1500 watt and a fan cited as providing noise-free forced-air operation. Model 105 is a compact unit working on 120 volt, 1320 watt.
Fasco Industries, Inc., Rochester 2, N. Y.

INJECTION-MOLDED TEES

Previously available in sizes from ¼ through six in., injection-molded tees are now offered for eight-in. unplasticized polyvinyl chloride (UPVC) piping. Recommended working pressure is 75 psi. Their use, with flanges and couplings injection-molded in the eight-in. sizes, makes feasible use of UPVC in large size piping.
Tube Turns Plastics, Inc., 2929 Magazine St., Louisville 11, Ky.

VERTICAL AIR FLOW CONDENSERS

Thirteen vertical air discharge models, ranging in capacity from 5 to 60 ton (with dual units available up to 120 ton), have been added to the Zephyrcon line of air-cooled condensers. In addition, the company has expanded its line of horizontal air flow Zephyrcons from 14 to 18 single-unit models. All units are equipped with a cross-fin coil with staggered copper tubes and aluminum fins. Dynamically balanced propeller fans operate at low speeds and each belt-driven condenser is equipped with two matched,



adjustable, standard V-belts. Designed for heavy-duty operation, motors are drip-proof and are located within the unit housing for protection from weather damage.
Larkin Coils, Inc., P. O. Box 1699, 519 Memorial Dr. S.E., Atlanta, Ga.

News highlights of the month

TRENDS

34,000 in 1964 Projected figures compiled by the Engineering Manpower Commission of Engineers Joint Council, 345 East 47th Street, New York, N. Y., indicate that by 1964, graduating classes of engineering colleges will total about 34,000. The present level is around 37,800. Because of natural attrition, a graduating group of 37,800 calls for a freshman enrollment of perhaps 70,000.

Engineers' salaries Based upon a survey of 24,000 engineers, the National Society of Professional Engineers, 2029 K Street, N.W., Washington, D. C., has issued a 54-page booklet "Professional Engineers' Income and Salary Survey – 1960" which indicates that, from the standpoint of salary, engineers continue to gain increased recognition for their work. Median income reached $10,660, an increase of 6.6% from that in a similar 1958 survey. Copies $2.00 each.

Engineers in industry National Society of Professional Engineers, 2029 K Street, N.W., Washington 6, D. C., announces "The Engineer in Industry in the 1960's – a Professional Program." Within the 148 pages of this study appear examinations in specific detail of a variety of present company programs serving to enhance the professional development of engineering employees.

Rockwell establishes Cited as the first corporation to take part in the newly organized Great Scholars Program of Pennsylvania State University, the Rockwell Manufacturing Company has established a distinguished professorship in engineering there. The program has the announced objectives of advancing "the standards of excellence in the fields of graduate teaching and basic research in science, technology and the humanities." Rockwell has pledged a minimum of $125,000 over a five-year interval.

More jobs for graduates Reporting upon postgraduate commitments for engineering graduates of 1961, the Engineering Manpower Commission of the Engineers Joint Council cites 91.8% had on June 15 either secured jobs, decided upon postgraduate studies, been committed to military service or had other plans. This is 11% higher than for business and commercial groups. This year, 14.3% engineers are entering graduate studies toward advanced degrees in engineering.

INDUSTRY CHANGES

Sweat cooling For proposed rocket nozzles there may be used sweat cooling, whereby liquid is forced into a nozzle through porous walls. Cooling is produced by the rapid evaporation of the fluid on surfaces in contact with the exhaust flame. Porosity would be varied from section to section of the nozzle so that greater cooling would be provided in the nozzle throat than in the divergent section.

BOOK REVIEWS

Cryogenics Covering the interval of 1954 to the present, the Cryogenics Engineering Laboratory, National Bureau of Standards, Boulder, Colo., has prepared a 16-page list of published material in its field. The Laboratory is equipped to provide processed copies of any of these 197 references which are no longer in direct supply. Prices vary according to a published list.

Absorption refrigeration Completion of a new study of the influence of chemical structure on such thermodynamic properties of absorption refrigerants as vapor pressure, heat capacity and activity coefficients of solutions is reported by the American Gas Association, 420 Lexington Avenue, New York 17, N. Y. Conducted by Professor Donald H. Andrews and Raymond A. Rousseau of Johns Hopkins the study is reported as "Solution Thermodynamics for Absorption Refrigeration." Price $1.50.

British handbook Now in its 10th Edition, the "Handbook of British Refrigeration Material" is a 272-page volume comprehensively reviewing facilities and progress. A 4-part section of 133 pages relates to machinery, power and plant equipment, insulation and refrigeration construction and instrumental products of the industry.

A technical section is focused on engineering details. Refrigeration Press, Ltd., Maclaren House, London, S.E. 1, England. 30 shillings.

ASTM in 1961 Annual Report of the American Society for Testing Materials, 1916 Race Street, Philadelphia 3, Pa., has just been issued as a 22-page Bulletin. Reviewed are technical articles, national and other meetings, publications, district and related activities, membership, honors and awards, fellowships and grants-in-aid, finances and various administration matters.

Heat insulation As published by the National Insulation Manufacturers Association, 441 Lexington Avenue, New York 17, N. Y., "How to Determine Economic Thickness of Insulation" is an expanded edition of a previously available manual. In its present 180-page form, this guide to the specification of heat insulations for flat surfaces and pipes at elevated temperatures is priced at $8.00.

Temperature measurement Design, construction and operation of various temperature measurement devices is the subject of "Temperature Measurement in Engineering, Vol. II", by H. Dean Baker, E. A. Ryder and N. H. Baker. Published by John Wiley & Sons, 440 Park Avenue South, New York 16, N. Y. This $13.00, 510-page is at the how-to-do-it level, as was predecessor Vol. I.

Building components National Academy of Sciences–National Research Council, through its Building Research Institute, 2101 Constitution Avenue, Washington 25, D. C., is distributing NAS-NRC Publication 911, Preassembled Building Components, containing the proceedings of the recent BRI conference.

SPECIAL MEETINGS

Engineering conference At the 1961 Cryogenic Engineering Conference, held August 15-17 at the University of Michigan, there were numerous technical papers keyed to such problems as space simulation, physical equilibria, future cryogenics, equipment, fluid transfer phenomena, cryogenic properties, heat transfer phenomena and mechanical properties and test procedures.

Cleaner Air Week As designated by a resolution of the 87th Congress (USA) October 22-28 will be Cleaner Air Week. Objective is to promote the public understanding necessary to evoke a countrywide approach to air pollution control. Nothing really new, Cleaner Air Week has earlier been endorsed by Presidents of the United States and proclaimed by more than half the governors of our 50 states.

Power conference At the 1961 National Power Conference in San Francisco, September 25-27, there will be sessions devoted to nuclear plants, boilers, high temperature steam, operating economics, automation and instrumentation. Sponsored by the American Society of Mechanical Engineers, there will be 23 formally prepared papers presented.

To honor Carrier There will be a joint ASHRAE-ASME Psychrometrics Meeting in New York on November 18 to honor the work of the late Willis H. Carrier (Presidential Member) and mark the 50th anniversary of his presentation of his Rational Psychrometric Formula before ASME. Special events, including a luncheon, technical discussions and announcement of the new ASHRAE Psychrometric Chart are planned.

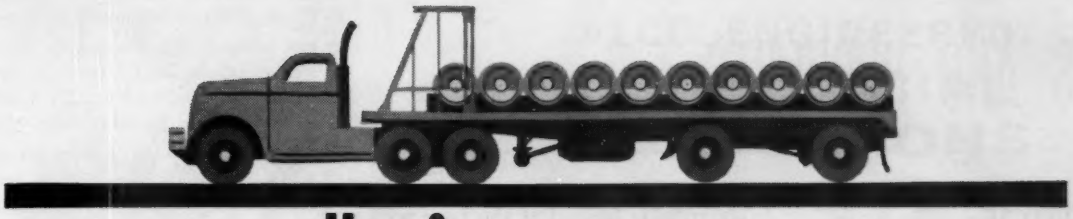
Food congress At the Fifth International Food Congress and Exhibition, New York, September 8-16, the theme will be "The Life Line of Humanity, Food From Farm To Table". First of these world-wide forums to have been held in the United States, this Congress will cover nutrition, marketing, communications, research and development, physical distribution, distribution operations and food and drug laws.

. . . WANTED . . .

A Technical Secretary

Because of an unavoidable, last minute, change of plans a vacancy exists for a Technical Secretary on the Headquarters Staff of the Society.

If you are interested in this position, or know of anyone who might be or who would be qualified, please contact the Executive Secretary.



Ucon® refrigerants RACK TRUCK

—for speedy, on-the-spot delivery of one-ton cylinders.

Ucon® refrigerants COMPARTMENTED TANK WAGON

—for one-trip delivery of two grades of refrigerants, each compartment having 15,000-lb. capacity.

Ucon® refrigerants TANK CAR

—for economical rail delivery in 6, 8 and 10-thousand gallon capacities.

THREE WAYS TO ROLL UP SAVINGS!

...with the most complete delivery facilities in the refrigerants industry!

These are the vehicles that deliver Ucon® Refrigerants to you in bulk, reducing your costs through quantity discounts and lower freight rates. Countless man-hours in unloading, testing and handling are eliminated. Whatever your refrigerant requirements, one of these service vehicles can quickly, dependably and *economically* meet them!

Let Carbide help you enjoy the economy and convenience of bulk shipments!

- Free technical assistance available to help you establish bulk storage facilities.
- Complete design and specification data for the bulk installation best suited to your requirements.

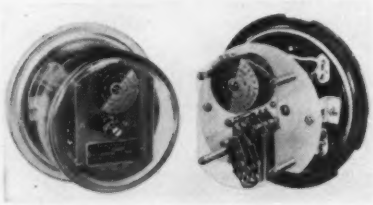
We draw on many years of experience in handling more than 100 chemicals in bulk quantities! Your local Ucon Refrigerants representative has the whole, eye-opening story on these bulk delivery facilities and the *big savings* they offer. Call him today...and put this money-saving fleet to work for you! UNION CARBIDE CHEMICALS COMPANY, Division of Union Carbide Corporation, 270 Park Avenue, New York 17, New York.

UNION CARBIDE CHEMICALS

PARTS AND PRODUCTS

TIME SWITCH

For staggered or sequence switching of heating and regeneration loads in off-peak stored hot water space heating installations, a new three-circuit time switch is offered. Provided is a



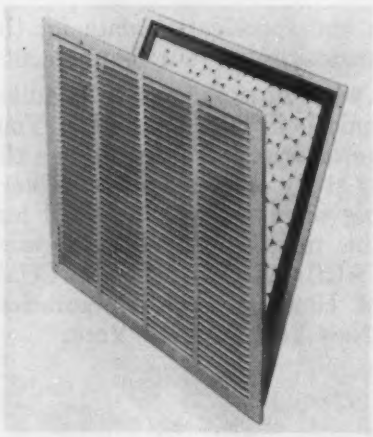
24-hr timing cycle, and the unit carries a stack of three dials, each with a set of on and off trippers. Through a positive locking linkage, each dial actuates a corresponding switch, allowing programming of the three circuits on individual schedules. Trippers are adjustable at 15-min increments. Switches are rated at ten amp, and the standard timing motor is supplied for use on 220/440-volt, 60-cycle current.

Basic construction is of non-corrosive brass, aluminum alloy and phenolics, with stainless steel shafting and springs. Non-operating steel parts, such as support brackets, posts and fasteners, are protected by zinc plated surfaces.

Reliance Time Controls, Inc., 1927 Mead St., Racine, Wisc.

FILTER-TYPE GRILLE

Suited for installations where heating and cooling systems are located in

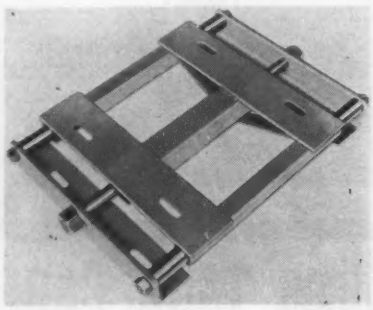


attics, crawl spaces and other inaccessible areas, this filter-type return air grille is hinged at the base with a full-length piano hinge and has a frame and filter rack allowing a one-in. filter to be installed directly behind the grille. Units are being produced in sizes to handle 500 to 2000 cfm. Half-in. louvers on the grilles are placed ½ in. apart and deflected 30 deg for maximum free area.

Lima Register Company, 1790 N. Cable Rd., Lima, Ohio.

MOTOR BASES

Three new series of all-welded, tension-controlling motor bases with freely movable carriages, designated Automatic, are cited as minimizing wear on belts and bearings by providing the proper amount of tension



even under heavy starting loads. Readjustment, when necessary, is made while operating under load merely by turning a screw.

Series 100 is available in standard and vertical types for 19 Nema frame sizes and has a self-contained spring to control belt tension and compensate for belt stretch. Designed for use with spring-loaded, variable-pitch pulleys, Series 200 is available in the same sizes as Series 100. On the BB Series, which accommodates larger motors, the motor carriage rides on linear ball bearings that extend the length of the carriage. Two standard and vertical models are available.

Manheim Manufacturing & Belting Company, Manheim, Pa.

COLOR-CODED FILTER

Three new features are cited for this hexachlorophene-treated, color-coded, glass fiber furnace filter. Hexachlorophene is provided to combat bacteria. The filtering medium is spun with progressing density, to provide more dirt-catching efficiency. Color-coding enables rapid recognition of the different sizes: 16 x 20 x 1-in. filters have green labels, 20 x 25 x 1-in. units have orange labels, red for 20 x 20 x 1-in. models, blue for the 16 x 25 x 1-in. filters and yellow labels for all other sizes.

American Air Filter Company, Inc., 215 Central Ave., Louisville 8, Ky.

AIR CURTAIN DOORS

As opposed to conventional doors, which must be opened and closed manually, a person enters or leaves a building equipped with Air Door through a moving current of warmed air. This downward-directed stream of air represents the only barrier between the inside and outside of the building, and is cited as reducing heated and conditioned air loss. Offered by this manufacturer are AirPac, a packaged product designed for commercial entrances four, six and eight ft wide, a modular unit, built in standard components for entrances up to 16 ft in widths and a custom product for buildings requiring even wider entrances.

Air Door, Inc., New Castle, Ind.

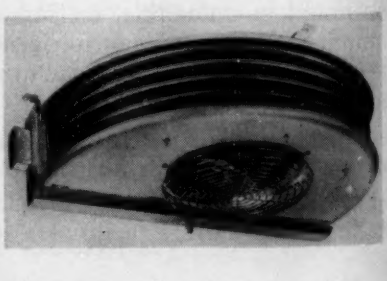
DOWNFLOW FURNACE

Added to the Compact furnace line, this down-flow design, gas-fired furnace is available in three sizes: 80,000, 100,000 and 120,000 Btu/hr. Measurement of the unit is 51 in. high, 26 in. deep and 16-3/16 or 20-3/16 in. wide, depending on capacities. Each size is offered in a straight heating model or in a high capacity air conditioning model, equipped with extra capacity blower and high velocity permanent filters.

Day & Night Manufacturing Company, P. O. Box 2222, La Puente, Calif.

UNIT COOLERS

Engineered to provide more head room and usable space in walk-in



coolers and refrigerated areas, the redesigned Half-Round Evaporator permits mounting directly against the back wall and ceiling. Air is drawn in from below and discharged radi-

ALCO'S 3 in 1 TK THERMO® VALVE

ALCO ENGINEERED COMMERCIAL APPLICATIONS

HI-TEMP...LO-TEMP.

Change capacity in 10 seconds on the job!

CAPACITY CHANGE DISCHARGE TUBE
.093 ORIFICE DIAMETER
.120 ORIFICE DIAMETER
New! FLAT WRENCH SUPER-HEAT ADJUSTMENT
ROUGHING IN DIMENSION 4½ x 2⁹⁄₁₆

REDUCES VALVE INVENTORY 3 to 1

2 discharge tubes—.093 orifice diameter and .120 orifice diameter are packaged with each ALCO TK THERMO VALVE for convenient "on the job" capacity changes.

CAPACITY CHANGES IN TONS

@ 100 F. CONDENSING — 40 F. EVAPORATION

VALVE No.	REFRIGERANT 12	VALVE No.	REFRIGERANT 22
TK 1 F	1.3	TK 2 H	2.2
WITH ORIFICE .120	0.6	WITH ORIFICE .120	1.0
WITH ORIFICE .093	0.3	WITH ORIFICE .093	0.5

THERMO VALVE SELECTION TABLES FURNISHED ON REQUEST.

SEE YOUR ALCO WHOLESALER

ALCO VALVE CO. 855 Kingsland Ave. • St. Louis 30, Mo.

BUY SECURITY • BUY QUALITY • BUY ALCO

The one complete line of refrigerant controls: Thermostatic Expansion Valves • Refrigerant Distributors • Solenoid Valves Refrigerant Filter-Driers • Suction Line Regulators • Flooded Evaporator Controls and Reversing Valves

8367

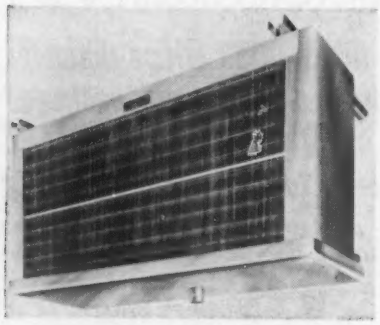
ally in a full 180-deg arc. The fan panel is hinged to a stationary drain trough; the drain line can be installed permanently, but there is still easy access for servicing the unit.

Other installation and service features have been incorporated in the unit, such as a system of slotted hangers for added ease in mounting and location of the electrical junction box on the outside of the cooler. Capacities range from 2600 to 14,900 Btu/hr at 10 F TD.

Bohn Aluminum & Brass Corporation, Danville Div, Danville, Ill.

CARCASS CHILLER

Designed for initial chilling of dressed carcasses, this new line features galvanized construction, totally enclosed motors and insulated drain pans. Units for Refrigerants 12 and 22 are equipped for water defrost. Ammonia units may be piped in the field for either water or hot gas defrost, or both. Capacities are to 75,000 Btu/hr

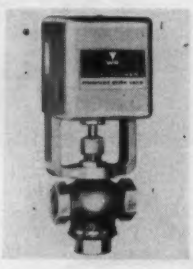


at 10 F TD for ammonia and 60,000 Btu/hr at 10 F TD for direct expansion units.

Recold Corporation, 7250 E. Slauson Ave., Los Angeles 22, Calif.

GLOBE VALVE

Addition of a heavy-duty, motorized globe valve to their line of hydronic zone controls has been announced by this company. While the valve's primary use will be providing zone control of steam or hot water heating systems, many other types of application are feasible: flow control of steam, water, oil or air; control of liquid levels; maintenance of desired pressure conditions; and automatic control of a complete process on a time or program basis.

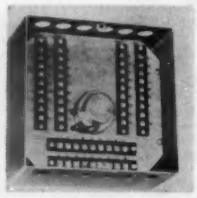


Powering the unit is a continuous duty-type motor of shaded pole design. All gears are machine-cut hardened steel. Non critical of mounting angle, the valve operates uniformly in any position not more than 90 deg from the vertical upright. Sizes available range from ½ to 3 in., either 32 or 139-sec timing cycle and in 24, 120 or 240-volt models.

White-Rodgers Company, 9797 Reavis Rd., St. Louis 23, Mo.

CONTROL PANEL

For use on any zone-controlled air or hydronic year-round heating and cooling system is a new Zone-A-Trol heating-cooling control panel and selector switch that permits changing from heating to cooling cycle and back again by setting the switch to the desired cycle.



When the switch is changed from heating to cooling, the motorized control panel automatically cuts the burner off and the compressor in, in addition to reversing the action of all thermostats so that zone controls open on temperature rise instead of temperature drop.

Econo Products Company Div, Viking Instruments, Inc., East Haddam, Conn.

RUBBER DOORS

Now available for use in refrigerated areas, flexible, self-closing rubber doors with positive air seals are offered by this manufacturer. These are basically auxiliary doors to serve where standard doors must be left open because of traffic activity. Units are constructed of duck-reinforced rubber with heavy-duty frame and have a self-returning cam hinge. In addition, auxiliary closing devices are provided. A choice is offered of ¼-in. diam wire, double-acting, adjustable compression springs or 5/16-in. diam wire, single-acting torsion springs for closure against heavy drafts.

W. B. McGuire Company, Inc., P. O. Box 265, Champlain, N. Y.

OIL-FIRED FURNACES

Comprising the new line of Weathermaker oil-fired furnaces are nineteen models of four major types: horizontal, upflow, counter-flow and Lo-Boy. Each unit has primary and secondary heat exchangers designed to extract extra heat from exhaust gases. Each insulated cabinet contains a heavy-gauge heat exchanger, stainless steel combustion chamber and large blowers. The air return can be on either side. Plenums which accept standard cooling coils are available for all units except the horizontal models, which can be matched with a new line of encased, horizontal coils. Seven horizontal models range in heating capacity from 85,000 to 335,000 Btu/hr; four vertical and three counterflow units are available at 85,000, 125,000 and 156,000 Btu/hr; and the five Lo-Boy models range from 100,000 to 250,000 Btu/hr.

Carrier Corporation, Carrier Air Conditioning Company Div, Syracuse 1, New York.

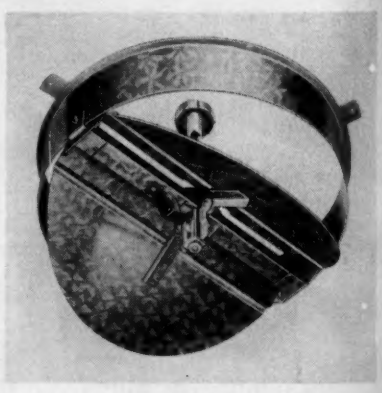
FIELD CLOSURE MATERIAL

Developed for making underground piping field connections, Rip-Coat is a mastic colloidal resin cited as being highly resistant to moisture, weathering, acids, alkalies and electrolysis. It is unaffected by temperatures ranging from sub-zero to 250 F. In the field, the material is brushed on over the welded conduit joint and then a glass-wrap material is applied over it. A final coating is applied over the wrap material to complete the closure.

Ric-wiL, Inc., Barberton, Ohio.

GALVANIZED DIFFUSERS

Changeover from cold-rolled to galvanized steel in construction of ceiling diffuser dampers and installation rings has been announced by this manufacturer. Improved resistance to rust and corrosion is cited, and the absence

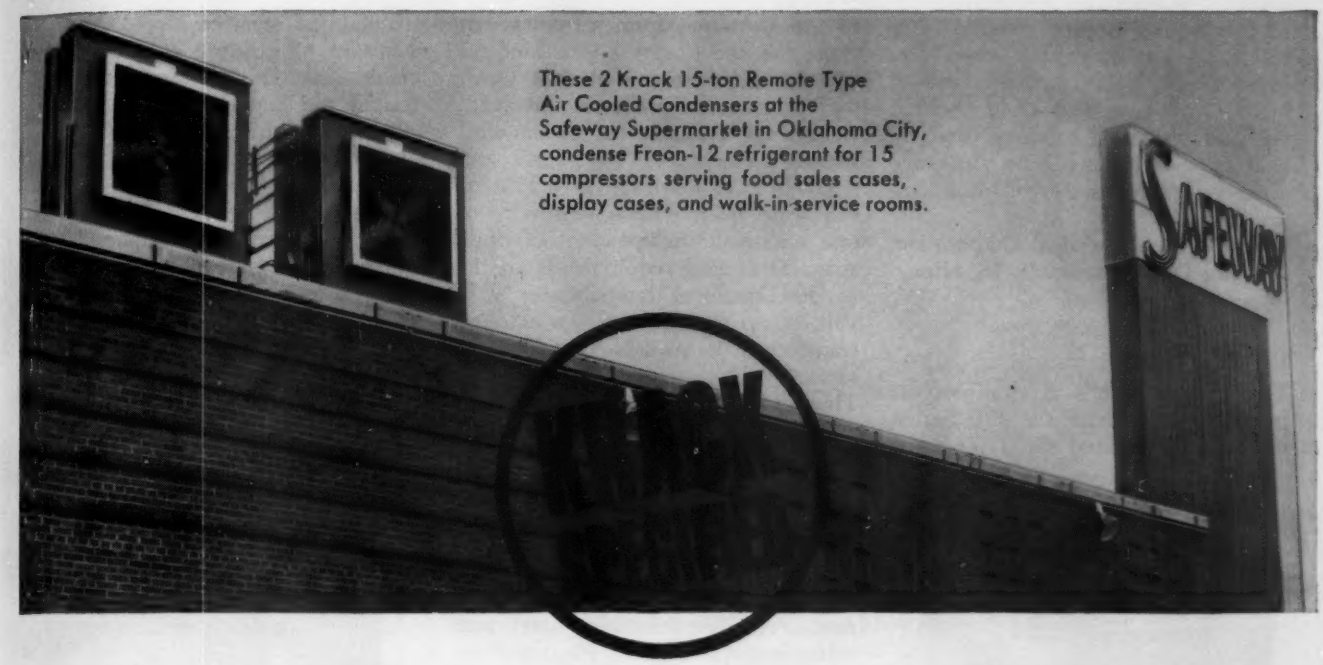


of paint around the operator mechanism insures a smooth opening and closing action. Dampers are operated by a screw-type nylon rack and pinion mechanism. Backlash tolerances in the unit are close to zero, in order to prevent blade flutter under high velocities.

Air Control Products, Inc., Coppersville, Mich.

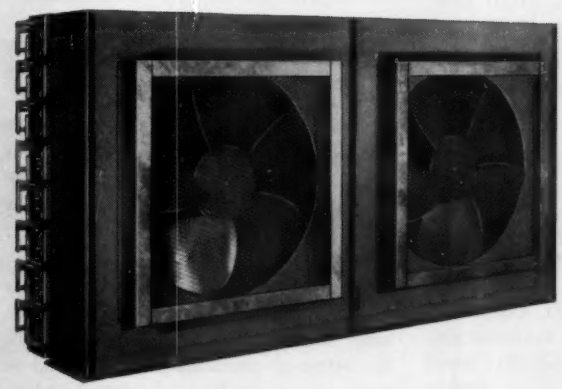
POLYETHER FOAMS

Two new rigid polyether foam systems have been added by this manufacturer



These 2 Krack 15-ton Remote Type Air Cooled Condensers at the Safeway Supermarket in Oklahoma City, condense Freon-12 refrigerant for 15 compressors serving food sales cases, display cases, and walk-in service rooms.

Count on KRACK to Deliver RATED CAPACITIES



Service calls? Not with Krack Air Cooled Condensers

Eliminate maintenance problems when you install Krack Air Cooled Condensers. Forget about down time for draining, scale removal and periodic overhauls. No water is required. Destructive corrosive action from water is gone and so are costly callbacks.

Krack Air Cooled Condensers give completely automatic performance all year 'round . . . deliver top capacity at any temperature. Krack Winter Control compensates for pressure variations caused by temperature changes. One push button stops or starts complete system. Multiple circuiting available to suit the individual job. Choice of direct drive or belt fans. Capacities: 1 to 90 tons.

"We chose Krack Air Cooled Condensers for Safeway because of full rated-capacity performance, low cost, and better delivery."

—L. C. Nettleship, Jones-Newby Supply Co., Oklahoma City

You can depend upon all Krack refrigeration and air conditioning equipment to consistently and safely perform at top capacities. You enjoy these advantages:

Be sure of matching the equipment to the job

With Krack there is no uncertainty about actual performance. You don't have to "play it safe" by specifying more costly, "oversize" equipment for a job when it is not needed.

Gain the advantage of lower bidding

By saving the cost of specifying "oversize" equipment to compensate for uncertain performance, you can bid lower and win more jobs.

Save installation and maintenance headaches

With Krack job-rated equipment, installations go more smoothly, and costly service calls for repairs and adjustments are rare. A good way to retain priceless customer good will.

KRACK REFRIGERATION APPLIANCES INC.

Manufacturers of Freon, Recirculated, Flooded or Direct Expansion Ammonia Heat Transfer Equipment

MAIL COUPON FOR FREE BULLETIN AC-457

REFRIGERATION APPLIANCES, Inc.
917 W. Lake St., Chicago 7, Ill.

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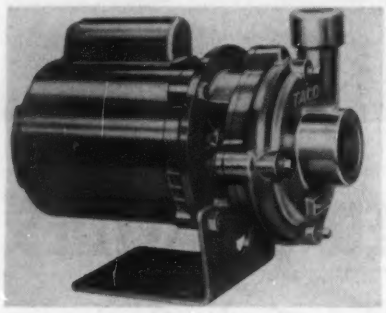
to the regular line of Pleogen foam resins. Major advantages cited are: low viscosity (1000 cp) for ease of mixing and flow; a built-in one-min delay for extended handling time; no heat cure is required; no shrinkage after cool-down; and excellent dimensional stability.
American Petrochemical Corporation, Mol-Rez Div, Minneapolis 18, Minn.

PUMP LINE

This manufacturer's general purpose pump line, formerly the 170 Series, has been designated the 1700 Series



and expanded to include twenty sizes with motors ranging from ¼ to 7½ hp. Compact, lightweight and easy to service, the units are suitable for many applications.
Taco Heaters, Inc., 1160 Cranston St., Cranston 9, R. I.

GLASS FIBER INSULATION

Made of high density glass fiber, a new thermal and acoustical insulating material is available to replace combustible materials in home air conditioners and other appliances. Normally supplied in sheets ⅛ in. thick, Thermo-Mat insulation is coated on one side against moisture penetration with a fire-resistant asbestos fiber mastic. Adhesives are applied directly to the uncoated side for installation.
Fibrous Glass Products, Inc., a Subsidiary of Pall Corporation, Alpa Plaza, Hicksville, N. Y.

MONITOR

Provided by this instrument is monitoring of from one to ten points, to be checked for the same value. Applications include monitoring temperature, pressure, speed, flow or any process variables measurable by transducers generating dc millivolt outputs. Each point to be monitored is scanned in sequence and its value compared to a pre-established set-point. Scanning speeds available are from one to ten sec per point. High or low deviation from set-point will cause the unit to give a visual indication of malfunction and shut down the process or processes involved. Ranges are available for all standard sensing elements.

Calibration accuracy is 0.25% of full scale span, with a sensitivity of one microvolt, independent of scale span. Operating requirements are 15 to 40 C ambient temperature; 115-volt, 50 to 60-cycle ac. An external transformer is available for 230-volt service.
Thermo Electric Company, Inc., Saddle Brook, N. J.

REFRIGERATORS

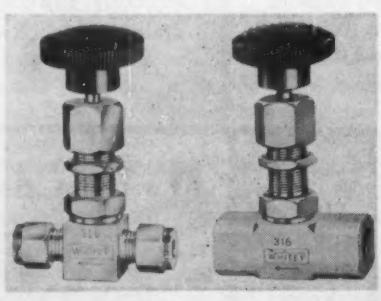
New Chill-Air Reach-In and Pass Through Refrigerators for storing meat are available in a number of models for supermarket, hotel and restaurant applications. For rapid assembly and installation, the units are built in prefabricated, knock-down sections. Standard models have two, three or four doors on one side; others have pass through doors. High density glass fiber insulation lines each refrigerator.
Erickson Industries, Inc., River Falls, Wisconsin.

THREE-WAY COIL

Introduced on this company's horizontal furnace line is a new coil case which will accept the same V-coil used on its upflow and downflow forced-air furnaces. This makes a single type of coil applicable for most air conditioning applications. Accommodated by the 48/60 DNH are 48,000 and 60,000-Btu/hr capacity coils. Introduction of cases for other sizes is planned for the future.
Day & Night Manufacturing Company, 855 Anaheim-Puente Rd., La Puente, Calif.

BAR STOCK VALVES

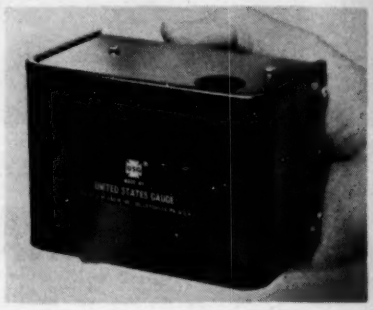
For ⅛ and ¼-in. tube and pipe connections, #2 Series screwed bonnet,



bar stock valves are available in type 316 stainless steel or Monel. With a choice of tube fitting connections or female pipe thread connections on straight and angle pattern styles, the valves can also be ordered with Swagelok inlets, female pipe outlets or female pipe inlets to Swagelok outlets.
Whitey Research Tool Company, 5525 Marshall St., Oakland 8, Calif.

SAFETY SWITCH

Designed for installations where bearings, packing glands and motor windings depend on pumped water for

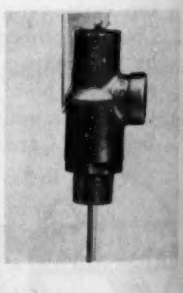


lubrication and cooling, this water level safety switch is cited as preventing pump damage due to insufficient liquid at the source, or when the pump loses its prime. Liquid flow is sensed by measurement of discharge pressure and the unit automatically shuts the system down when the water level gets too low.

Mounting on the pump discharge line, the switch is connected, in series, with the pump motor power supply. If, after a predetermined period of time, pump discharge is not at a specified pressure level, a microswitch electrically disconnects the pump motor, stopping the system.
American Machine & Metals, Inc., United States Gauge Div., Sellersville, Pa.

RELIEF VALVE

For hot water storage tanks, this valve automatically opens when water temperature reaches 205 F or pressure exceeds 125 psi, to prevent explosions. Pressure relieving capacity of the unit is 125,000 Btu/hr. Thermal unit of the valve extends six in. into the water.



Capitol Manufacturing Company, Buckeye Valve and Manufacturing Company Div, 415 S. 18th St., Columbus 5, Ohio.

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Comment

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OFF WITH THE OLD; ON WITH THE NEW

It is probable that, by the time this is published, your Society Headquarters will have been established in the United Engineering Center, 47th Street (345 East) and First Avenue, in New York City. Vacated will have been the old ASHAE offices at 62 Worth Street as well as those of predecessor ASRE at 234 Fifth Avenue. We shall be joined at UEC by the staffs of 18 other professional engineering societies.

Much has been written and said as to what the new Headquarters can and will mean to the members; some of it to engender interest as well as to impart information, some of it to whip up contributions to the Building Fund itself. All of it a tribute to a truly outstanding concept reflective of engineering unity and presumably singleness of broad purpose.

For our part, we would emphasize the basic concept that lies behind that oneness. ". . . we are a professional engineering society", as then-President Bob Tull reminded in his talk before the Denver 68th Annual Meeting "not a trade association, not a promotional group, not a social club".

We are engineers.

Progressing to our new Headquarters might well be a rededication to the nobler aspects of professional engineering society membership, activities and purposes.

BIGGER AND BETTER

Entering upon a sustained program to increase the membership of a Society already impressively large, as reported upon elsewhere in this issue, is an ambitious—and sobering, undertaking.

Just how large should a Society be in relationship to fields of activity, foreseeable imminent growth, attritions, paralleling developments in related fields, and all the other imponderable factors?

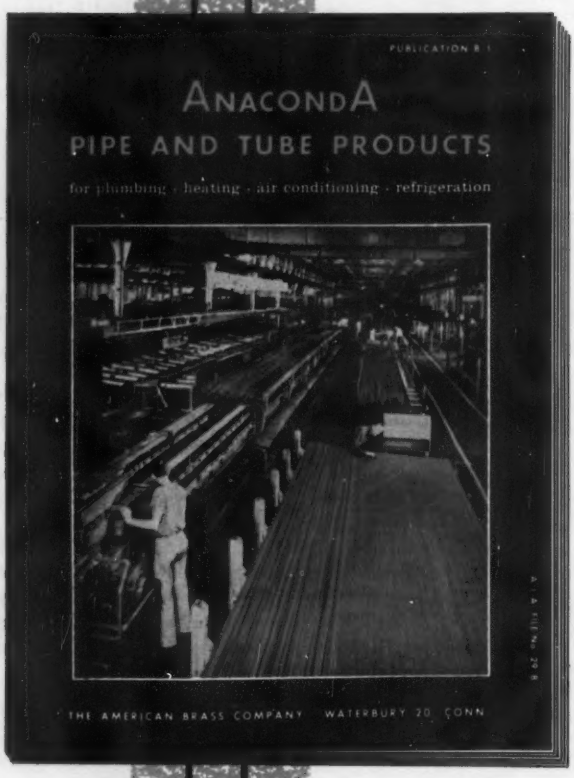
ASHRAE is headed for a relatively modest goal that represents a net growth of perhaps 5% a year.

We look forward to its successful outcome.

RIGHT QUOTE; WRONG AUTHOR

To the late Boake Carter are due this observer's regrets. He was the one to whom we should have credited "In time of war, the first casualty is truth" as quoted on this page in our July issue.

Edward R Searles
Editor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How we designed an

HTW Central Heating Plant

for the Montreal Airport

To serve the Montreal Airport with a central heating plant required finding a solution that would take into consideration various previously installed heating systems. Indeed, there were over 35 individual heating units in approximately 20 different locations. A survey of loads showed a maximum present requirement of 160 million Btu/hr; the forseeable future development is expected to increase the load to 250 million Btu/hr. It further was established that the present summer load would amount to approximately 10 million Btu/hr.

A detailed analysis of the data indicated that, if a central heating plant were to prove an economical replacement for the individual plants, it would be either high temperature water in the 370 to 400 F range, or 125-psig steam. Preliminary plans of both the plant and the distribution system were prepared for both cases and budget estimates prepared. From these estimates it was established that the high temperature water system would be approximately the same in initial capital cost but have an annual owning and operating cost at least 15% less than with steam. These savings arose out of reduced costs in water treatment and maintenance and reduced losses in blow-down, flash losses and trap losses.

Previous studies on similar installations by our firm had shown that, with the heat storing characteristics available with high temperature water when the load has

D. LORNE LINDSAY
Member ASHRAE

daily fluctuations, it is possible to use this factor in reducing the peak output of the heating units. From an estimated load curve and knowing the heat storage capacity of the distribution system, it was established that the peak capacity of the "boilers" could be reduced by 12% should high temperature water be used. This factor, of course, contributed largely to the attractiveness of hot water.

Largely due to expensive coal and ash handling and storage costs and a competitive price for oil, the Coal Board allowed the use of bunker oil as the primary fuel when the savings were shown to be approximately 22% in this case. Shortly after the beginning of construction of the plant, the local gas supplier offered a new rate structure based on an interruptible supply clause, and the plant is now equipped to burn both oil and natural gas.

Once the budget estimates had been completed and owning and operating costs could be estimated accurately, it was established that an attractive annual saving could be made if the existing individual heating plants could be abandoned and replaced by the central system—this notwithstanding the fact that the existing plants were largely written off.

The estimated annual saving was $17,000 at present and, of course, as new buildings were added to the system, the figure could be expected to increase greatly. The total budget for the entire project was just under $3,000,000 and now that the project is complete, we are pleased to report that the final cost has not exceeded this figure. On the basis of a few months operation and known operating costs, it has recently been established that heat can be sold profitably to commercial users at a rate of $1.25 per million Btu.

After having considered several possible locations, the site for the plant was chosen remote from the area of the Air Terminal, but in the vicinity of the second largest load center of the administration and industrial buildings area. The plant was built with a central core 60x75 ft in plan and about 55 ft high. The basement is 14 ft high. An extension 16 ft with glazed walls runs on the front and on two sides of the central core and contains the pump room, diesel generator room, offices, lunchroom, locker-room, washrooms and workshop. The central core contains four boilers and the expansion control tank on the ground floor. All piping, as well as such auxiliaries as forced draft fans, air compressors, water make-up and water treatment plant, fuel oil pumping and heating plant and pressure tank are in the basement.

D. Lorne Lindsay is a Consulting Engineer with Wiggs Walford Frost and Lindsay in Montreal. This is a condensed version of a paper presented by the author at the American Power Conference held in Chicago, March 21-23, 1961.

In planning the design for a central heating plant to service Montreal Airport, the author's firm had to answer such questions as —

What load should the plant be capable of handling?

What future development of load might be expected?

Was one plant and distribution system the most economical solution?

What heating medium should be used?

What was the most suitable fuel to burn?

What might be the total cost of the plant to suit present and immediate future requirements?

On a platform above the boilers are located induced draft fans, expansion storage and make-up tank, feed water deaerator and air supply units for combustion. Access to all levels in the plant is provided by means of stairs and platform.

Distribution System—The piping tunnels start from the basement of the plant. Where more than two large size pipes are running in parallel, a walk-in tunnel is provided. Where the building of tunnels was found uneconomical due to small size pipes, the distribution piping was installed in insulating concrete conduits.

The external distribution system has been divided into four separate circuits. Each circuit is comprised of a pair of pipes, one starting at the 16-in. main supply header, the other at the 16-in. main return header in the basement of the plant.

Boilers — Four high temperature water boilers were installed, two of them with a continuous output of 70 million Btu/hr, two with a continuous output of 40 million Btu/hr. Each boiler has been specified to be able to take a one-hr peak load of 10% over the continuous output capacity. Boiler tests recently completed show satisfactory operation at 120% of specified capacity.

The boilers are of a non-orificed forced circulation type with the circulation augmented by gravity circulation through external "downcomers". With this arrangement the heat absorption of each tube is compensated properly, resulting in a unit capable of operation at low flows without circulation problems and having a low pressure drop on the water side at specified full flow.

The return water enters a header at the base of the rear wall and then through the convection section into a header at the top of the boiler. The front and side walls are fed from the convection section discharge header through external downcomers to their bottom headers, up through the water walls to top headers which feed into the main supply header. Additional external downcomers connect the front and side bottom headers to the top headers and these tubes are so sized that the effects of gravity circulation are substantially greater than the water friction anticipated at maximum flow conditions. Thus, any tube absorbing a greater proportion of heat, which would result in a higher average temperature, will, through gravity circulation, increase its flow to compensate its greater heat absorption. With this feature the problems of orifice sizing for widely varying loads such as are characteristic of heating system loads are overcome.

The gas flow within the boilers is so arranged by baffles that the gas flow is counter to the water flow. With the counter flow arrangement, exit gas temperatures are quite low, being about 100 F higher than the entering water temperature. In the recent acceptance tests it was shown that with 260 F return temperature and 400 F leaving temperature on the water side, an exit flue gas temperature of 380 F was obtained, giving an overall efficiency of slightly over 85%.

Circulation—Water circulation in the plant and in the system is maintained by means of three high temperature water circulating pumps. Space has been allowed for the future addition of a fourth pump. Pumps are of the vertical design with cast steel casings and stainless steel impellers. The pump glands and bearings are water cooled. Each pump has a capacity of 1500 Imperial gpm against a head of 180 ft water gauge, or 72½ psi at 280 F. Pumps are located on a depressed platform below the ground floor level, with access from the ground floor and from the basement as well. One of the pumps has been provided with a variable speed motor of unusual design. Control of the pump speed provides means of adjusting pump capacity and head to the momentary requirements of the load and this pump will be mainly used in low load conditions. Pump speed can be controlled either manually from the control panel or automatically to maintain constant pressure differential in the equipment room of the new Terminal Building. This point was chosen as being characteristic for the whole system and it is assumed that when the pressure differential is sufficient in the new Terminal Building, it will also be sufficient in all other buildings.

The temperature in each of the four circuits leaving the boiler plant can be controlled automatically or manually by means of mixing the boiler outlet water with the system return water. The amount of heat used in each circuit is recorded and integrated continually. In buildings where it was found necessary, the use of heat is metered by means of individual Btu meters.

Independently from the external circuits, high temperature water also is used to heat the Central Heating Plant, to heat the fuel oil, to deaerate make-up water and other such services. In some cases, instead of using the supply water at 400 F, the return water (which will normally be about 260 F) is used.

The distribution system has been sized for a total circulation of

approximately 4500 Imperial gpm. At present, with all four boilers in operation, a circulation of approximately 3000 Imperial gpm can be maintained. Of the available head of 180 ft in full load conditions, 80 ft will be lost in the distribution system, 50 ft in the Central Heating Plant and the remainder in building equipment rooms. A relatively high residual head was provided at building entrances. This made easier the task of sizing control valves and also contributed to reducing the size of piping in the new Air Terminal where long distribution lines were run from the Equipment Room in the vault to the wings and aeroquays.

SYSTEM CHARACTERISTICS

The system was built for high temperature water being generated at 400 F. The corresponding saturation pressure is 233 psi approximately, but the system operating pressures are higher since the system is being pressurized by means of nitrogen and also the pump head adds to the total system pressure. The design pressure of the system is 350 lb and the main components of the system were built for this pressure. A margin of not less than 15 psi over the saturation pressure is maintained automatically in any point of the system in any operating conditions.

A vertical tank 5 ft diam and 12 ft high, located on the ground floor, together with a horizontal tank 5 ft diam and 12 ft 6 in. long, located in the basement, provide space for water expansion and for control of the system pressures. The combined volume of the two tanks is 490 cu ft, or over 3000 Imperial gallons. The tanks are branched on the circulating pump suction header by means of an 8-in. connection. The pressure and level controls in the tanks are automatic and arranged so as to maintain the system pressure in a linear relationship to water level in the upper vertical tank. When water expands in the system, the water level in the tank rises and compresses nitrogen. The pressure everywhere in the system will rise and should it exceed the safe limit, the safety valve will release some nitrogen from the top of the tank to return the pressure to the safe maximum. However, this is not likely to happen in normal operating conditions, since when the upper level in the tank reaches 12 in. above the normal water level, a level control valve starts to modulate between closed and fully open, passing greater or lesser quantities of water to the expansion and make-up tank upstairs. The level in the expansion control tank is thus maintained constant at this upper limit.

Should this fail, an alarm will signal the operator the abnormal condition when water level reaches 13 in. above normal. When water in the system contracts, the water level in the tank drops and the pressure of nitrogen is reduced. When the pressure of nitrogen in the expansion tank reaches the safe minimum, an automatic pneumatic control valve will admit some nitrogen to maintain the system pressure at a safe minimum. This situation, however, can only happen should some nitrogen be lost previously, due to some leakage in the system, or should the safety valve be allowed to pop. In normal operating conditions before it happens, water which has previously been stored in the expansion and make-up tank when overflow valve has operated, will now be returned to the system by means of transfer or feed pumps.

There are two transfer pumps and one feed pump installed in the plant, and when the water level in the tank reaches 11 in. below normal, a transfer pump of 20 Imperial gpm capacity starts transferring water from the make-up tank to the system. Should this not be sufficient, another pump — this one 50 Imperial gpm capacity — will start and an alarm will be sounded, when water level in the expansion control tank drops to 13 in. below normal. In case of some emergency condition when the water level drops to 24 in. below normal, the largest feed pump will start, this one having a capacity of 100 Imperial gpm. Pumps will cut off automatically, the feed pump when level has returned to 12 in. below normal, the No. 1 transfer pump when level has returned to 9 in. below normal and the No. 2 transfer pump when water reaches the normal level.

Except in extremely varying load conditions, water level in the expansion control tank will remain within the limits set by the level at which No. 1 transfer pump cuts in and the level at which the overflow valve starts to modulate. The volume of expansion or contraction which can be accommodated between these limits amounts to 235 Imperial gallons. A reserve capacity of water is stored in the bottom of the tank and it amounts to 340 Imperial gallons.

For this purpose a reserve capacity of water is stored in the bottom of the tank and it amounts to 340 Imperial gallons. This amount serves to prevent emptying the tank and sending nitrogen into the system in case some difficulty were encountered in obtaining an immediate start of pumps and it will provide the operator with the time required to undertake the necessary emergency action.

WATER MAKE-UP SYSTEM

A centralized system of make-up water treatment and distribution was designed to serve the high temperature water system and the equipment rooms in most buildings where steam generators are used. The centralized make-up supply and treatment plant originates from a city water main. Since the pressure in the main is fluctuating and may not be sufficient to overcome all resistances encountered in make-up system, a booster pump, designated the first stage booster pump, was installed at the city water entrance to the Central Heating Plant. This pump is sized to deliver 100 Imperial gpm when increasing the supply pressure by 20 psi. When pressure in the city main is high enough, water flows through a by-pass around the pump. When pressure drops below the set minimum (50 psig) a pressure switch will start the pump. Also to assure a positive water supply to the deaerator any time the supply valve starts to open, the pump will be started by means of a switch mounted on the lever of the deaerator make-up water controller.

Treated water may contain an excessive amount of free oxygen and, therefore, it may be directed to a deaerating heater. The make-up system was sized for emergency

conditions to provide up to 100 Imperial gpm of treated water. However, in normal operating conditions the amount of make-up water required by the high and low pressure systems is but a small fraction of the above capacity. It was, therefore, considered excessive to provide a deaerator of the emergency capacity, especially when the unit would require extremely oversized heating facilities.

A heater sized to deaerate 100 Imperial gpm of water to the standard oxygen content of 0.05 ml per litre would have prohibitive size and weight and a considerable amount of heat would be lost when little water would flow. Taking the above into consideration, the deaerator was sized to reduce oxygen in 37 Imperial gpm of soft water at 40 F to 0.05 ml per litre. The deaerator is of the spray type and water is heated to 228 F at 5-psig steam pressure. In normal operating conditions when deaerated water demand is lower than the design capacity of the heater, the amount of oxygen left in water will be less than predicted for the design conditions. In emergency conditions of high feed water demand, the deaerator will be bypassed. Sodium sulphite, of which an appropriate concentration is maintained in the high temperature water, will then neutralize any oxygen that may be introduced.

Water used to feed steam generators in equipment rooms may be treated further if and as required, by adding appropriate chemicals by means of a chemical feed pump. The same pump can introduce sodium sulphite into the high temperature water circuit for oxygen treatment and also sodium phosphate to maintain the pH value of the high temperature water at 9 to 10. A concentration of 30 to 40 ppm of sodium sulphite usually is maintained in high temperature water.

Softened, dealkalized and, if necessary, deaerated water is used to make up any water loss in the high temperature water system. The make-up water is introduced into the expansion and make-up tank located on the upper platform.

The expansion tank is pressurized with nitrogen up to 50 psig but treated water in the deaerating heater is under 5-psig pressure only. It was, therefore, necessary to provide second stage booster pumps. They were installed in the basement of the Plant and they are boosting the pressure to enable it to enter the tank. Two second stage pumps were provided and installed side by side with the first stage booster pump. They have a capacity of 100 Imperial gallons each and they increase the pressure of water coming from deaerator by 60 psi. Pumps are operated alternately by means of an alternating switch and are controlled automatically from a switch on the lever of the make-up valve of the expansion and make-up tank when water is required in the tank or else from a flow switch in the line to the equipment rooms when water is required to make up the steam generators there.

Most of the present users of the high temperature water in this heating system are using the high temperature water to generate low or high pressure steam. The resultant return water temperature might be higher than would be desired for proper and economical operation of boilers. Since several services in the Central Heating Plant do not require truly high temperature water, it was decided to use for these services water which returns from the distribution system, instead of using 400 F water as it leaves boilers. To achieve this, the return water pressure must be boosted so as to provide enough head to overcome friction in the service heating system. This system serves the space heating of the Central Heating Plant, the combustion air heating, the domestic hot water heating and all heating requirements of the heavy fuel oil.

Piping Systems – Besides the piping system for high temperature water distribution, piping was installed in Central Heating Plant, in tunnels and in equipment rooms for distribution of cold and hot utility, feed and make-up water, for auxiliary heating system, for space heating in Central Heating Plant and in equipment rooms, for boiler blow-down, for fuel oil and gas distribution and for compressed air distribution. A drainage system was provided also for the Central Plant and for tunnels.

Pipes in tunnels and in the Central Heating Plant are supported on sliding supports and guided where required. Anchors are arranged so as to avoid excessive heat loss from pipes. Pipe anchors have two ribs welded to the pipes and a plate bolted to the structural frame. A ¾-in. asbestos insulating plate was inserted between the anchor plate and the rack for thermal insulation. Sliding supports are free to move on lubricated metal plates supported from the structural racks and the pipe is supported by means of a formed insulating concrete block so that minimum heat will be lost while proper movement of the pipe will be secured. Guides are made similar to supports but rollers were added on the sides to secure straight line pipe movement.

HTW Water Piping – In the basement of the Central Heating Plant the four circuits originate from the two main 16-in. diam headers, the supply header and the return header. Air cylinder operated parallel slide type valves in the supply and return pipes of each circuit can be operated from the main control panel on the firing floor of the Plant to shut off any circuit.

Return water is screened in passing through a 16-in. strainer; it then passes through a mixer where supply water at 400 F can be added to raise the return water temperature to the minimum required for safe boiler feed. A control valve in the supply water line to mixer can be operated from a manual positioner on the main control panel.

From the extended return header, water passes into the pump suction header and by means of one or more of the three circulating pumps it is pumped into the pump discharge header. Air cylinder operated, 10-in. globe valves in pump discharge connections can be operated from the main control panel to eliminate pumps not in operation. Valve opening can be adjusted manually to throttle pump discharge if required. Check valves are also provided in discharge connections.

From the pump discharge header two branches feed return

water to one side of the Plant where one small and one large boiler are located. Each branch can be shut off by means of an air cylinder operated valve from the main control panel. At each boiler inlet a check valve and an air cylinder powered parallel slide shut-off valve were installed. The shut-off valve can be operated by means of a switch on the respective boiler panel. A similar remotely operated shut-off valve can be found at each boiler outlet side by side with a butterfly valve for boiler water flow adjustment. The butterfly valves also are operated from the respective boiler panels by means of pneumatic positioning relays.

A by-pass was arranged from the circulating pumps discharge header with connections to all four supply lines at their origin at the supply header. Each connection is equipped with a pneumatic diaphragm operated control valve. These control valves will admit return water to the supply lines of the distribution circuits to maintain on each circuit the desired temperature lower than the boiler outlet temperature. This arrangement will be used in low load conditions to save on line heat losses and to increase the water flow in the system should it fall below the safe boiler flow rate. The operator will have the choice of either using this control or the reheating by-pass to control the flow.

Equipment Rooms and Building Systems – At present, the only buildings in which high temperature water is being used directly for space heating are in the new hangar for Canadian Pacific Airlines and the aeroquays and wings of the new Air Terminal.

In the Canadian Pacific Airlines hangar high temperature water is circulated through projection type unit heaters to provide heat for the main body of the hangar. On the way back to the Central Heating Plant, water is cooled further in baseboard fin tubes in offices. The distribution to the unit heaters is of the reverse return type and water flow to the two branches of the system is controlled from the outdoor temperature by means of a single modulating control valve. Each branch has been equipped with a two position three-way control valve which will divert water flow from the fin tubes when heat is not required in offices. Unit heaters will be operated intermittently as called for by room thermostats operating unit heater motors.

In the wings and in aeroquays of the Air Terminal high temperature water is distributed to several air heating units of the ventilating and air conditioning system. On load decrease, water flow to the coils is throttled by means of a modulating control valve and some water is diverted to the return line by means of another similar valve in a by-pass connection. This provision was made to maintain the supply water at high temperature ready for prompt air heating any time the heat is called for after a prolonged shut-down period.

At the Air Terminal in an underground vault three steam generators were installed, each sized to produce 16,000 lb/hr of dry saturated steam at 75-psig pressure from water at 180 F. If necessary, steam pressure can be raised to 100 psig but the capacity of each generator will then be reduced to 10,500 lb/hr. Steam is used in air heating coils throughout the building and in kitchens. High temperature water is used to heat domestic hot water to provide heat for low pressure hot water heating system and also to heat antifreeze in the snow melting system. The heat exchangers for these services are located in the basement of the main body of the Air Terminal Building.

The vaporizers in the vault are supplied with condensate from the main building. On its way back to the vault the condensate passes through a heating coil in a water storage tank to preheat water which then is heated further to the final temperature in the domestic hot water tanks by means of high temperature water. This arrangement saves flashing condensate to steam. Condensate collected in a tank in the basement then is transferred to a condensate tank in the vault, from where it is pumped to the vaporizer. If necessary, treated water from the water treatment plant in the Central Heating Plant is added to the condensate tank on request of a level controller. Thermohydraulic feed water regulators control the distribution of feed water to the three vaporizers. One of the two feed water pumps is in continuous operation.

Heat to the Air Terminal Building is delivered by means of two circuits, 10-in. diam No. 1 circuit and 6-in. diam No. 2 circuit. The two are interconnected in headers in vault from where water is distributed to several services. Heat used in the building is metered in each circuit separately by means of liquid meter type Btu meters.

FUEL OIL SYSTEM

The fuel oil system was arranged to provide three kinds of service:

To supply hot heavy oil to burners

To supply Diesel oil to burners in emergency

To supply Diesel oil to the emergency Diesel-generator

The oil storage tank farm is arranged to receive oil by truck loads

Filling connections are arranged at a shed near the tank farm and oil is pumped from there to either one of the two heavy oil tanks or to the Diesel oil tank.

The temperature of the oil in the tanks is maintained at or above 90 F by means of bottom heaters. The control of temperature is manual by means of needle valves in the hot water returns from the bottom coils. Since oil is delivered to tanks at a temperature much above the required 90 F, a bottom heater will be used only when a tank has been standing by for a prolonged period of low temperature weather.

The oil pumped from a tank to the Plant is heated in a tubular heater to the desired pumping temperature (around 100 F) and the temperature is controlled automatically by means of a pneumatic temperature control valve in hot water supply to the heater. The valve modulates water flow to maintain oil temperature at the set point in suction line.

Piping from tanks runs in an insulated trough to the outside of the dyke where it enters a shed covering the outlet of the tunnel leading to the fuel oil pump room in basement of the Plant. All piping in the trough and in the tunnel

is insulated, except Diesel oil carrying pipes. All heavy oil pipes are traced with hot water tracers. Water flow in tracers is controlled by hand by means of needle valves.

The oil pumping and heating station is arranged for two stage pumping of heavy oil with preheating of oil between stages and for single stage Diesel oil pumping to burners. A valve in the light oil supply to pumps is operated electrically and interlocked with a similar valve in hot water supply to heaters to prevent possible heating of Diesel oil should heater be left operating by mistake.

In the first stage oil is taken from one of the storage tanks to one of the pumps, pumped to the first stage pump discharge header and further to a PRV which maintains a constant pressure at oil inlet to the heater. This pressure is kept at a value somewhat lower than water pressure in the heater tubular heating coil. Heated oil is mixed with the return oil from burners and sent to the second stage suction header and to a second pump. This pump pumps oil to a pressure which will result in the desired pressure at burners.

CONTROLS AND INSTRUMENTATION

Operation of the whole district heating system is based on a variety of automatic or semi-automatic control systems. In general, these control systems are intended to control the flame, heat generation, heat utilization, heat distribution and the pressure and water volume in the system.

Since oil is heated by means of coils supplied with hot water, the control of oil temperature is similar to the control of all other heat exchangers and consists of the temperature sensing and transmitting instrument and a pneumatic diaphragm operated valve with valve positioner. The control has a large proportional band and reset and rate adjustment and also a throttling valve in air supply to the diaphragm. This type of control provides all the necessary means for adjusting the actual performance of controls to the design requirements. Oil pressure is maintained constant in different stages of oil pumping by means of pressure relief valves.

The intent for the combustion control system is to provide the best coordination of all elements of the high temperature water generating equipment at all loads and firing rates and for any number of generators in operation. The combustion control system shall assure the highest generator efficiency, shall protect generators against any dangerous operating conditions and shall provide operators with means of instantaneous observation of all elements of the operation while continuously adjusting the rate of heat input to the changing water flow rate and water inlet temperature, in order to always maintain the generator outlet water temperature at the precise desired value.

The rate of firing is being adjusted to the instantaneous load in accordance with the signal representing the combined signals coming from the temperature differential transmitter and from water flow temperature re-corrected to the signal from the actual outlet water temperature. The desired rate of firing is maintained by means of parallel control of combustion air and fuel flow with automatic re-correction of fuel flow to maintain the desired relationship between measured fuel flow and measured air flow.

Final control of fuel flow is accomplished by means of an oil flow control valve in the oil return line or by a gas flow control valve. Both control valves are equipped with positioning relays with characterizing cams. A solenoid valve in the control air line will transfer the control signal as desired from a transfer switch in the burned programming and flame safeguard equipment.

Final control of air flow is accomplished by means of air vanes on forced draft fan inlet. The vanes are positioned by means of an air operated piston which is controlled from a relay with characterizing cams. A signal from the programming controller will offset the combustion controls and, as required, either fully open the air vanes for the generator purge or it will send the controls into low fire starting position. This is required when starting the first burner on a generator. Low fire start position will be confirmed back to the programmer by closing appropriate switches in valve and vane operators.

The high temperature water circulation system is of the single pump type. The motion of water is therefore due to the action of the same pump (or set of pumps working in parallel) whether it is moving through generators or through the distribution system. The rate of flow through generators is, in a system like that, directly dependent on the rate of flow at users. Since most controls at users are of the direct flow control type, the changes in heat requirements at users are almost directly reflected in the amount of water circulating through generators. The sole difference is the insignificant amount of water passing through by-passes provided at ends of lines and in equipment rooms. The operator has means either to increase the flow through generators in relation to the flow through the distribution system or else to increase the flow through the distribution system without increasing the rate of flow at users while either passing the additional flow through the generators or not, as desired.

Heat distribution controls are of two basic groups. One is formed by automatic controls at users, which are controlling the delivery of heat subject to the instantaneous load demand at each point where heat is consumed. The other group is intended to adjust the conditions of temperature and available head in the system to changing load conditions so that the controls of the first group will not be subject to extreme pressure variations in the characteristics of the controlled medium. This group of controls also provides the operator with the means of operating the Plant to some extent independently of what are the instantaneous load requirements at users.

Controls at users are controlling the amount of heat transmitted from the primary heating medium, which is the high temperature water to the secondary medium, in which different parts of this system are the low or high pressure steam, low pressure hot water, domestic hot water, ventilating air, environment air or anti-

(*Continued on page* 83)

Design factors in

Better Laboratory Hood Design

Adequate laboratory hood design calls for meeting certain basic engineering requirements, based on the premise that the unit must control contamination of the specific location through the use of ventilation.

Speakers at ASHRAE's Industrial Ventilation Symposium at the Denver Annual Meeting, June 26-28, 1961, covered the subject in depth; from design to choice of structural material; from venting problems to cost considerations.

What primary requirements are needed in controlling contamination through ventilation?

J. C. BURKE, JR.
Associate ASHRAE

Among first considerations in laboratory hood design is the nature of the contaminant to be controlled. Is it mildly undesirable or extremely hazardous? Is it in a gaseous, vapor or particulate form? In normal laboratory work, overall requirements would probably be confined to heat, fumes and dust, therefore, the hood must vary depending on this contaminant.

If heat only is involved, most of the exhaust will be expelled near the top of the hood. If the contaminant is heavy, or generated near the hood floor, then the exhaust location will be near the bottom. If the operation is concerned with highly toxic or corrosive contaminants generated in small quantities, a secondary hood or localized exhaust pipe may be utilized to permit lower air quantities and, possibly, the use of lower cost materials for the hood construction.

Often the big consideration is the first cost of the hood. For most any laboratory contaminant, we can run the full gauntlet of hood types. We can put a nozzle type of suction opening near the source of contamination, and attempt to drag it all the way by this means. We can provide a hooded enclosure with an open face, in which the man works in the open face area. We can provide a nearly complete enclosure, or one that is so complete that the operator must work through gloves installed in the walls of the hood.

First cost may not be a primary consideration because of the process involved. For instance, a highly contaminating material might be contained effectively only in the totally enclosed hood, in which case first cost is secondary.

The totally enclosed hood will, however, get by with less air, which brings us to the consideration of air to feed the hood. A prevalent habit of many organizations is to install an excess of hoods without worrying from where the air is coming to supply the hoods. If we are drawing air out of the structure through hoods, the air has somehow to get into the building.

It is desirable to have inflow into the laboratories because with it one tends to have better control over contaminants. If air is provided properly for the hoods, one has to consider the cost of air supply systems, and include this as an item of hood overall capital expense.

Next is the cost of treating the exit air. Hood operations may require air incinerators for treating infectious bacteria, or highly efficient filtration may be necessary to keep exhaust air pollution to a safe level.

Expense entailed would include not only air treatment but also air handling facilities, exhaust fans, duct work, the whole building exhaust system.

Another highly significant point in air handling expenses imposed by improper hood selection is use factor. In laboratories utilizing many different hooded operations which, because of the processes, may be used only occasionally, but must be available immediately, the situation can exist where several hoods are attended by one operator. Obviously, one operator cannot use all these hoods at one time,

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J. C. Burke, Jr., is head of the Mechanical Engineering Department, Tracy-Behrent. The paper excerpted above was presented as "Basic Requirements of Laboratory Hoods" at the Industrial Ventilation Symposium at the ASHRAE Annual Meeting in Denver, June 26-28, 1961.

nor can he be relied upon to conserve air by keeping all unused hood doors closed.

An additional expense with regard to large air handling systems is cost of maintenance. All equipment from the inlet air filtration, through the heating and cooling equipment, to the final stages of exhaust air treatment has to be maintained. Another basic requirement affecting laboratory hood design is hood location.

Where is it going to be? Is this hood going to be placed in a room that already has a half dozen hoods? Is use factor going to be such that the amount of equipment to be enclosed means that every wall of the room is covered completely with hood face? Location then, becomes a quite definite limitation on what one can do with the hood. Location near a door, near a large air supply outlet, or near a number of other hoods definitely will affect performance because of possible cross drafts tending to upset the face velocity barrier.

Consider in the design stages the use of materials from which the hood is constructed. The contaminant handled in the hood may determine the hood material or, as is often the case, the solution or solvent used to clean the inside of the hood to prevent contaminant build-up may be highly corrosive and the cleaning agent then would necessitate the use of stainless steel for the hood construction. The totally enclosed hoods may require stainless steel construction for strength and durability as well as for corrosion. What then are we going to use for our hood construction?

The essential nature of a laboratory hood necessitates the use of air velocity. This velocity is across the face in open hoods or is a potential air velocity achieved by a negative pressure in the case of totally enclosed hoods. It must be the protective barrier between the operator and the contaminant. The magnitude of this velocity is, of course, dependent upon the construction of the hood and its intended use and must be kept to a minimum in order to conserve air. This face barrier might be designed to achieve a specific velocity and to include extremely smooth or uniform flow over the entire face of the hood.

Some of the problems the hood, as designed, must face when it is put in service include filling with operating equipment; hot plates, blenders, mixers, glass laboratory equipment and trees of apparatus growing to the limits of the hood capacity. Any convenient free space offers a possibility of mounting all types of thermalcouple gages, blending and supply equipment, for the apparatus in the hood. All in turn produce disturbances to the smooth flow of air into the face of the hood.

Important in the use of the hood is maintenance of air balance in the building and in the air handling systems. Air supply must match the exhaust at all times; otherwise there will be deleterious effects on the face barrier.

How do we determine the extent of hood containment? One way is by measuring face velocity for uniformity. Many organizations say that the specific face velocity that has been selected shall be maintained $\pm 10\%$. This means dispersion of exhaust openings, equipment out of the way and, for uniform face velocity, a clear area. Uniformity for which this velocity is checked is most important in deciding whether the hood is maintaining its intended operation correctly.

Another criterion often offered as absolute is maintenance of a specific magnitude of face velocity. Based on the criteria that are made available at the time of the design, let us assume a specific velocity is selected. Is this velocity that which actually is going to do the job?

It has been stated that a hood, in order to be effective, has to provide uniformity of flow and absence of turbulence of any kind in its interior. Methods of checking the hoods consist of using the velocity meter to check the face velocity. This indicates whether we have uniform face velocity and whether we are maintaining a specific velocity under all conditions. However, aren't we often missing the whole point of why we have the hood? The hood is intended to contain a contaminant. It is supposed to capture it; it is supposed to hold it within the enclosure and remove it from the area. Why not then, as an evaluation method, use a means of measuring the escape of contaminants from the hood? To evaluate a hood adequately, one has to measure the amount of material that is escaping under actual operating conditions – not when the hood rests under ideal conditions. If so, do we have to provide a greater enclosure? This would be the case if one were continuing to pick up contaminating materials around the laboratory. In other words, put more physical metal walls between the man and the contaminate and then have but one air face that has to be maintained to protect the man. If the contaminant persists in the room, more doors on the hood are provided or it is closed further. As you try to keep contaminants out of the space, you might eventually go all the way to the complete enclosure, but at any rate you are measuring your effectiveness by containment.

There are several different approaches to this problem of hooding and the various air flow problems connected with hood operations. The more hazardous and more demanding industrial hooding requirements have inspired development of the air foil entry hood and the unconditioned air supply hood.

The air foil entry hood is designed to provide smooth flow conditions at the hood face. This is accomplished by doing away with corner posts and the sharp edged counter tops through substitution of a rounded streamlined entry at the hood face.

The unconditioned air supply hood may or may not have an air foil entrance, but it has the feature of using a small amount of conditioned room air, supplemented by a separate supply of tempered outside air. Conservation of conditioned room air results in certain economic advantages for this type of hood in large air consuming applications.

Another modification that may be applied to hoods in an effort to help maintain air balance within the building and its ventilation system utilizes a by-pass arranged so that air enters the by-pass when the hood face is closed. This provides a more constant face velocity regardless of the sash front position.

One type of hood that has been used successfully in hooding applications has a sharp edged orifice entry. When air is induced into an opening the formation of a vena contracta results. This is accompanied by conversion of static pressure to velocity pressure and back to static pressure again. Around the vena contracta exists an area of turbulence and high negative pressure. Tests in this type of hood have shown that turbulence and eddy currents do exist but that these eddy currents assume a definite flow pattern toward the interior of the hood. Tests also have shown uniformity of face velocity and, of course, higher pressure drops, but most significantly, tests together with actual use have shown effectiveness against the escape of contaminants. Another interesting observation derived from experiments conducted with this sharp edged hood was the aspect of a volume – face area ratio.

The volume – face area ratio is a number obtained by dividing hood volume by hood face area – for instance if a twenty-cu ft hood had a face 2½ ft high by 4 ft wide, or 10 sq ft, then the volume-face area ratio would be 20/10 or 2. Effectiveness of the sharp edged hood increases as volume-face area ratio increased. In other words, for a hood of one specific volume, the smaller the face area, the more effective the containment at a constant face velocity.

In satisfactorily fulfilling laboratory hood requirements, the engineer must first determine the duty expected of the hood – the nature of the contaminant, the work done with the contaminant, the use factor and the proposed location of the hood. He must then make certain that the hood complies with certain basic requirements: 1. Compatibility with the building air handling systems, both supply and exhaust; 2. An adequate face velocity barrier; 3. Hood operator protection, safety, comfort, and convenience; 4. Proper air flow within the hood; 5. Suitable construction materials.

After these requirements have been considered in detail, and an economic evaluation has been made, the overall operation of the hood must be analyzed to assure that it fulfills its one basic purpose – to "satisfactorily control contamination."

Laboratory hood size, material and construction features are based on specific application

TOM B. LANAHAN

Application, in a way, is the easiest part of hood design, because one speaks of the purpose for which the fume hood is going to be used. Under application, one can define the type of hood required by asking the following questions: Is it to be used in a general research laboratory? Is it going to be used in the field of medicine and biology? Is it going to be used in the field of pilot plant operation?

This, in a way, will define the material to be used. For example, if the work to be accomplished is in the field of radioisotopes, or medical or biological research, stainless steel is desirable. If it is a general type laboratory fume hood, transite, colorlith materials are desirable. If it is an industrial type application, and possibly where the budget is limited, a carbon steel, acid-resistant painted hood, or even a wood hood lined with transite or colorlith can be used. Limit use only in areas for dry powder type chemistry where no possible fire will ever occur. In some applications for industrial use, where the chances of using fire are limited, and it is a production type chemical process, metal painted with an acid-resistant coating, with the work surface of alberene stone, colorlith, transite or stainless steel is practical.

The size of the fume hood also depends on the application. If one is to do pilot plant work, and knows his equipment approximates 16 ft long by 6 or 8 ft high, this would determine the size of the hood.

Materials selected for fume hood construction have to be predicated on the use and application of the unit. For example, if one is doing radioisotope research, or medical and biological research, there are stainless steel hoods with round corner construction available from a number of different companies. Material cost of the hood represents a relatively small percentage of the selling price; therefore, one should quite seldom take second best in the choice of material. The main cost in fume hoods is in the labor to manufacture them.

If one is going into industrial type research for the chemical or petroleum industry, a transite, colorlith or alberene stone hood is certainly desirable.

Group 1 is the conventional laboratory carbon steel type fume hood, painted with an acid resistant paint or baked enamel finish. The baked synthetic enamel finish is resistant to the action of the following reagents applied as a one-cc puddle of each reagent to the surface for a period of one hour:

Phosphoric Acid, 75%	Ethyl Alcohol
Sulfuric Acid, 25%	Methyl Alcohol
Nitric Acid, 25%	Acetone
Glacial Acetic Acid	Ethyl Acetate
Sodium Hydroxide, 10%	Ethyl Ether
Sodium Hydroxide, 25%	Chloroform
Ammonium Hydroxide, Conc.	Carbon Tetrachloride
Hydrogen Peroxide, 5%	Xylol
Phenol, 85%	Furfural
Formaldehyde, 37%	

Group II is colorlith, which I have used merely as a type of fume hood readily available. There are others which carry other trade names:

Hydrochloric Acid (Dilute–20%)
Hydrochloric Acid (Concentrated–36%)
Sulphuric Acid (Dilute–20%)
Sulphuric Acid (Concentrated–50%)
Nitric Acid (Dilute–20%) — Discolored Surface

T. B. Lanahan is Chief Engineer for S. Blickman, Inc. His paper titled "Material and Construction Features in Laboratory Hood Design" was presented at the Industrial Ventilation Symposium at the ASHRAE Annual Meeting in Denver, Colo., June 26-28, 1961.

Nitric Acid (Concentrated–50%)	
Phosphoric Acid	
Sodium Hydroxide	
Ether	
Naphtha	
Gasoline	
Acetone	No Visible Stain
Methyl Alcohol	
Ethyl Alcohol	
Benzene	
Toluene	
Xylene	
Carbon Tetrachloride	

Group III would be various grades of stainless steel alloy:
(a) 302, 304, 304L
(b) 316, 316L, 317
(c) 430
(d) 410, 420

The materials submitted above may be of help in suggesting the material used for hood design.

Starting out with the first type of construction in hood design, which is used quite seldom, is wood with a fire resistant liner such as transite or colorlith. In some applications, this type of equipment is perfectly justified as far as the end use in the choice of material. It can be put together by connecting with screws. To improve on the design, one can use either cement or epoxy to eliminate cracks or crevices. The third method would be a rosan type fastener. This is a metal insert with internal and external threads. Again, to improve on this design, one would use cement or epoxy filling at all joining surfaces to eliminate crevices.

In joining and fastening materials such as transite and colorlith, the wood screw principle is practicable, the rosan type fastener can be used, or the spline if the material is thick enough. If the material is thin, it then has to be fastened to a carbon steel back structural member, and the exterior mild steel painted with an acid-resistant paint.

If hoods are to be fabricated of carbon steel or stainless steel in which the interior and exterior of the hood are of the same metal, construction features such as spot welding, tack welding, continuous welding, would predicate the cost of hood fabrication.

The construction method of spot welding connecting surfaces is used when the hood will be painted with an acid resistant paint. The acid-resistant paint serves two purposes: it protects the surface and it seals crevices and cracks. Another method is welding, again on carbon steel where an acid paint coating is going to be used.

In connection with stainless steel fume hood construction, it is desirable to weld continuously the connecting surfaces. If the welding is done automatically and the interior of the hood is, for example, a 2B surface, a wire brush over the weld area is adequate. However, if the interior of a fume hood is to be polished to a #4 finish, then it is desirable to grind and polish the weld area so it will match and blend in with the adjoining sheets.

In the fabrication of fume hoods, a quite important feature after the material is chosen is not to overdesign established requirements. For example, if one selects wood and merely coats it with acid-resistant paint, one should maintain the criterion that this is a cheap hood. One should not hesitate to dispose of this hood upon damage to it, or when the criterion changes, another hood should be substituted.

A major error is to begin with a criterion of building a cheap hood, such as wood, and then over-engineering the unit. The hood should not be merely wood, it should be plywood; it should not be simply plywood, but marine plywood; it should not be put together with nails, it should be assembled with screws; rather than using screws, rosan fittings should be utilized.

This indicates clearly how one can over-engineer on both material and fabrication details. One may find out that it would be actually cheaper to buy a unit entirely of stainless steel with all round cornered construction. An important thing that one should look for is that the unit is as economically constructed as possible. For example, if one is interested in a wooden fume hood with acid resistant paint and a fire resistant liner, one should go to a wood fabricator. If one is looking for a stainless steel fume hood, one should go to a stainless steel fabricator, and not a wood fabricator.

In the specification of fabrication of hoods, it is always desirable to specify gauges of metals. It is not always true that a 4, 5, 6, or 8-ft hood can be built using the same gauge metals. If a hood is reinforced properly by use of hemming, or box channel formations, or even a box-corner type construction, this can lend itself to obtaining the rigidity required and make it possible to use lighter gauge metal. The most important areas to inspect for rigidity are in the post members where the window slides occur.

There is an excellent location for obtaining rigidity of the hood. It is most important that rigidity be obtained on the front post members and across the top front hood enclosure so that the window movement can be smooth, free and easy. It is even desirable when spot welded construction is used to utilize a right-angle hem bend, for example, around the top area of the hood.

To improve rigidity further, it is desirable to weld continuously all seals in connection with the internal baffles; it is also quite important to use proper gauges, and equally important to use right angle bending, hemming and diagonal creasing, so there would be no possible oil canning or vibration on the interior of the hood. This obviously can be eliminated by proper rigidity.

If a fume hood is to be used for radioactive type work or if lead caves are going to be placed on the interior of the hood, it is certainly desirable that the base support cabinet be reinforced with additional angle members to insure rigidity in the base cabinet, as well as putting a subway grating between the base cabinet and the fume hood on top. There may be occasions wherein a 48-in. unit, the stand or base cabinet will be required to support a 2000-lb load, or a 6-ft unit to support a 4000-lb load.

Heat and Mass Transfer

in dehumidifying surface coils

W. L. BRYAN
Member ASHRAE

In an air conditioning heat exchanger, such as a cooling coil, both heat and water are removed from the air. The added complication of vapor diffusion together with the transfer of heat confronts the refrigeration and air-conditioning engineer with special problems. The engineer has been faced with finding a proper method for rating and making performance calculations for the heat and mass transferred. Much research and analysis has been devoted to this problem and yet the fundamental basis for the methods now in use is debatable. In an attempt to achieve simplicity in application, assumptions have been made or empirical equations proposed. Aside from the loss in accuracy, especially if the operation conditions differ from the conventional range, the basis for the assumptions has not been fully justified.

Each of the commonly used methods, Contact Mixture Analogy, Carrier,[1] and Humidity Method, Tuve and Seigel,[2] assumes that the "Lewis Relation," Lewis,[3] applies. Lewis stated that the coefficient of heat transfer divided by the coefficient of vapor diffusion through a gas film is constant and equal to the humid specific heat of the gas. This relation was derived for the case of unsaturated air flowing over a free water surface and does apply with accuracy in the case of proper wet-bulb determination as shown by Carrier[4] and others,[5] but the assumption that this relation applies to dehumidifying coils has not proven correct.[6] One is justified in assuming a Lewis Relation at one only on the basis of simplicity with an accepted loss in accuracy. The relatively more complicated enthalpy potential methods[7,8] also are based on a Lewis Relation equal to one.

Based on the assumption that the Lewis Relation applies, the surface temperature of the coil for heat transfer is obtained by calculation. Although this temperature may be specified as a hypothetical value, it must be considered an approximation to the actual surface temperature required in determining the temperature of the refrigerant inside the coil. For a deep, extended surface coil or one operating at moderate load ratios, the accuracy is usually sufficient for engineering purposes. In the case of bare tube coils or shallow coils, the actual surface temperature may be greatly different from the calculated value.

In this first report on the cooperative research project, experimental results are given for the transfer of mass and heat by a bare surface coil under conditions of measured surface temperature. Experimental data are given for the heat and mass transfer coefficients and a simple fundamental equation is given for the actual coil surface temperature.

EXPERIMENTAL APPARATUS

A line diagram of the experimental apparatus is shown in Fig. 1. The apparatus was located on the balcony of the Mechanical Engineering Laboratory, which was large and well ventilated, so that recirculated laboratory air could be used. The intake air was humidified and heated before the fan and metered after a duct run. The sensible and latent condition of the air was determined before and after the coil by wet-bulb and dry-bulb thermometers and dry thermocouples. Care was taken to mix the air after the coil and check the leaving air condition after a set of straighteners. The air was metered again before discharge to the room.

Considerable work was done on the calibration of the temperature and flow measuring equipment. Orifices and nozzles used to meter the flow were checked by combination and pitot tube traverses and all flow pressure measurements were made with micromanometers. Matched mercury in

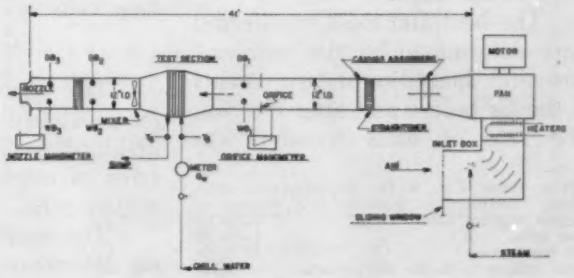

Fig. 1 Diagram of the experimental apparatus

W. L. Bryan is Associate Professor of Heat and Power, Mechanical Engineering Dept., Case Institute of Technology. This paper, the result of research sponsored by ASHRAE in cooperation with the Case Institute of Technology, was presented at the ASHRAE 68th Annual Meeting in Denver, Colo., June 26-28, 1961.

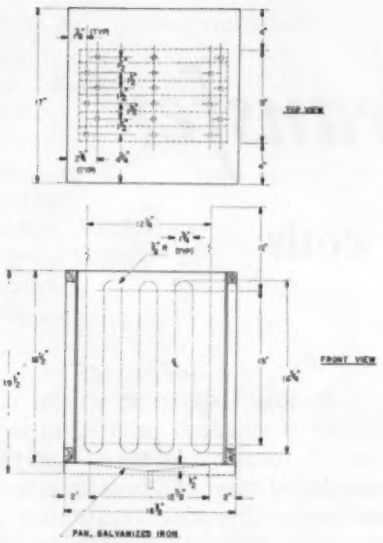

Fig. 2 Experimental test section for six-row bare coil

glass thermometers of 0.1 F accuracy and copper-constantan thermocouples through a type K potentiometer with an electronic null indicator were used to measure temperature.

The six experimental coils in staggered arrangement formed the test section, giving a six-row deep staggered bare coil, Fig. 2.* Each coil was insulated electrically and supplied through rubber connectors with cooling water from a supply header and return header such that the water flowed in parallel through each coil. With a high rate of water flow, the refrigerant temperature could be maintained substantially constant throughout the coils.

The actual wall temperature of the individual tube coils was measured by using each coil as a resistance thermometer in a Kelvin bridge circuit. The bridge circuit read the tube resistance to ½ micro-ohm for a resistance-temperature ratio of 4 micro-ohm per degree F for a tube surface temperature accuracy of better than 0.2 F. The temperature of any one of the six rows was determined independently of the others.

The heat and mass transferred were determined by the temperature and humidity measurements of the air before and after the coil. The rate of mass transfer was checked also by measuring the rate of condensate from the coil.

* Data under Fig. 2 for experimental coil:

Tubing 0.500 in. O.D., 0.450 in. I.D., copper	
Outside surface area	8.826 ft²
Face area	1.911 ft²
Net flow area	1.421 ft²
Ratio outside to inside surface area	1.111

RESULTS

Tabulated data and results of the experimental tests on the six-row bare coil are given in Table I. The surface temperature of each coil row was determined separately; however, the variation from row to row was usually less than ½ F so an average surface temperature for the total coil is given in the table. The sensible heat transferred and the latent heat, or mass transfer, tabulated are mean experimental values calculated from the data by enthalpy balance, temperature change and condensate measurements.

Since the refrigerant and thus the tube surface temperatures were substantially constant throughout the coil, the heat transfer coefficient was determined by using a logarithmic-mean temperature difference between the entering and leaving air and the mean surface temperature of the coil.

The heat transfer coefficient is defined:

$$dQ_s = h\,(DB - t_s)\,dA \qquad (1)$$

and

$$dQ_s = M_s\,c_h\,dDB \qquad (2)$$

by combining an integration, the coefficient for the total coil is:

$$\frac{hA}{M_s\,c_h} = \ln \frac{DB_1 - t_s}{DB_2 - t_s} \qquad (3)$$

The coefficient of mass transfer is defined in an analogous manner with a driving potential characteristic of the mass-transfer process. There being no agreement on the best parameter for the mass potential, the most convenient, the humidity ratio difference, was used. The mass transfer coefficient is defined by:

$$\frac{h_D A}{M_s} = \ln \frac{W_1 - W_s}{W_2 - W_s} \qquad (4)$$

The coefficients of heat transfer for these data were correlated statistically as a function of the coil face velocity giving the equation:

$$h = 0.616\ V^{0.5} \qquad (5)$$

The experimental heat transfer coefficient for the data is specified by Equation 5 with a standard error of estimate of 0.650 or it is within ±6%.

The coefficient of mass transfer determined in a like manner as a function of the face velocity is:

$$h_D = 1.313\ V^{0.6} \qquad (6)$$

The mass transfer determination was not as accurate as the heat transfer due to the more difficult measurements. Equation 6 gives the experimental mass transfer coefficient within ±10%.

It is of interest to consider the value of the Lewis Relation resulting from the experimental data. The Lewis Relation is

$$\frac{h}{h_D\,c_h} = \text{constant} \qquad (7)$$

The data for this bare coil give a ratio of heat to mass as a function of the coil face velocity. Using the mean value for the humid specific heat for all data at 0.25 and Equations 5 and 6 gives Lewis Relation:

$$\frac{h}{h_D\,c_h} = 1.877\ V^{-0.1} \qquad (8)$$

The use of a mean humid specific heat for any given point of data introduces an error of less than 3%. Equation 8 represents the bulk of the data within 10%. Some points of data, however, fall outside this range, since errors occurring in the coefficients in opposite directions cause additive error in calculating the value of the Lewis Relation.

The state of the air at any point, x, in the coil can be determined by the following equation when the surface temperature is constant:

$$DB_x = t_s + (DB_1 - t_s) \times \left(\frac{W_x - W_s}{W_1 - W_s}\right)^{\left(\frac{h}{h_D\,c_h}\right)} \qquad (9)$$

The derivation of this equation, similar to Bull,[9] is given in the Appendix. The exponent of the term for the water vapor content is the Lewis Relation (L.R.). If the L.R. equals one, the locus of the air states through the coil is a straight line. When L.R. is greater than one, the locus of states is a convex curve crossing the saturation line before reaching the surface temperature. For L.R. less than one, the resulting process curve is concave and always below a straight line connecting the initial air state and the coil surface temperature on the saturation line.

The experimental data presented here for the bare coil give a L.R. always greater than one. The L.R. would be one at a face velocity of 540 fpm for this coil. The data range covered was from approximately 125 to 450 fpm, which gave a L.R. range of 1.16 to 1.02.

It is important to notice that a L.R. of one for a bare coil is not probable at air velocities below 500 fpm. An air velocity of approximately 1000 fpm is required for L.R. of one for small diameter single cylinders allowed to transfer heat by both radiation and convection. The proper operating point for sling psychrometers is, therefore, around 1000 fpm.[10] The radiation shielding in aspirating psychrometers[11] permits lower operating air velocities.

SURFACE TEMPERATURE

The difference between the actual coil surface temperature and the hypothetical surface temperature used in the contact-mixture principle (by-pass factor method) or the humidity-ratio method should be evident. If the Lewis Relation is one, the actual temperature at the surface of the water film on the coil is the same as the surface temperature given by the contact-mixture principle proposed by W. H. Carrier. This would occur at one operating velocity. At other velocities two errors enter coil calculations based on either of these two methods. The calculation of coil tests from the air side data alone results in a hypothetical surface temperature. The calculation of the heat transfer coefficient from this hypothetical temperature gives a fictitious value.

A graphical picture of this error in temperature is shown in Fig. 3, where the data from a representative run, No. 2, are plotted on a psychrometric chart. The entering, leaving and surface temperature states are plotted. The locus of the leaving conditions for a similar coil of increasing depth is plotted also. When the coil becomes infinite in depth the leaving state will be the coil surface temperature. The surface temperature given by the contact-mixture method is approximately 60 F and that by the humidity ratio method 62 F.

Given sufficient surface in depth it is noted that the air is cooled beyond the saturation line or into a "fogged" state. This condition is likely at low velocities and high Lewis numbers as approached in natural convection cooling. The phenomenon of fog dropping from a refrigerant suction line on pumpdown represents this condition.

COEFFICIENTS

A plot of the experimental coefficients of heat transfer is given in Fig. 4. The solid line is the curve given from Equation 5.

Tests were made for this coil under conditions of sensible heating and cooling without dehumidification. The dotted line on Fig. 4 represents these data. The values for the dry-coil coefficients correspond to similar data by others, for example, Kays.[12] Since the air turbulence is greater during dehumidification, it was expected that

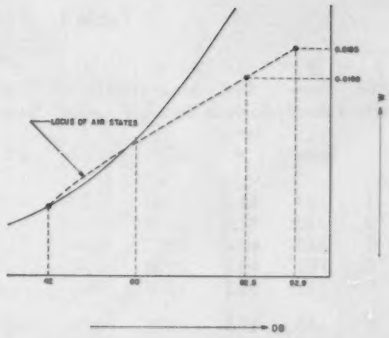

Fig. 3 Psychrometric chart plot for run No. 2 entering 92.9 F D.B. leaving 82.5 F D.B. and 42 F surface temperature

the heat transfer coefficient for dry-cooling would be lower.

If the heat transfer coefficient had been calculated without knowledge of the coil surface temperature curve "A," Fig. 4 would have resulted. This is an approximate calculation of the coefficient by the contact mixture principle. This curve crosses the experimental curve at a Lewis Relation of one, approximately 540-fpm air velocity.

The mass transfer coefficient is given in Fig. 5 and the solid line represents the statistical average curve for the data, Equation 6.

In considering these data for analysis it was found more accurate and convenient to work with the summary equations. These equations represent the best statistical correlation of the data for heat and mass transfer. The correlation of the total data showed a high degree of accuracy, although the difficulties of accurately meas-

Fig. 4 Experimental coefficients of heat transfer vs. coil face velocity

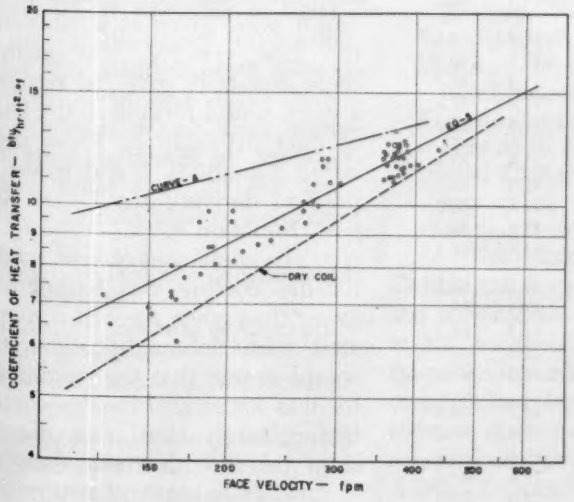

Fig. 5 Experimental coefficients of mass transfer vs. coil face velocity

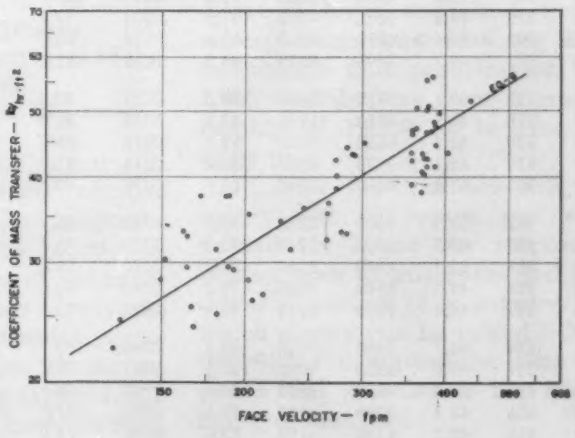

Table I Data and Results

1 Run No.	2 Face Velocity fpm	3 Coil Surface Temp. F	4 Heat Transferred Sensible Btu/hr	5 Heat Transferred Latent Btu/hr	6 Entering Air Temp. DB_1	7 Entering Air Humidity Ratio W_1	8 Leaving Air Temp. DB_2	9 Leaving Air Humidity Ratio W_2	10 Coefficients Heat h	11 Coefficients Mass h_D
1	127	43.2	2690	2430	99.9	.0185	83.0	.0162	7.2	22.6
2	130	42.0	2620	2690	92.9	.0185	82.5	.0160	6.5	24.8
3	150	44.4	2900	4540	97.2	.0240	87.3	.0206	6.9	28.3
4	152	44.1	2840	4620	96.7	.0242	87.1	.0205	6.7	30.3
5	163	55.2	2370	4930	96.6	.0271	88.2	.0234	7.3	33.3
6	164	55.5	2490	4690	94.8	.0257	87.1	.0224	8.0	32.9
7	164	43.4	3230	5540	99.6	.0260	89.6	.0237	7.2	29.5
8	166	43.3	2130	5620	98.3	.0254	88.7	.0237	7.1	25.9
9	167	42.0	2770	2760	90.2	.0157	81.7	.0141	6.1	24.1
10	172	47.0	2570	1360	88.0	.0130	80.5	.0117	7.7	37.7
11	181	70.0	2620	2270	105.5	.0240	97.9	.0225	9.4	31.3
12	182	42.9	2890	1970	89.0	.0148	80.8	.0135	7.8	25.2
13	187	74.3	2310	1360	104.4	.0229	97.9	.0219	9.8	37.7
14	189	43.2	3390	4790	92.3	.0247	83.0	.0216	8.6	29.7
15	190	42.9	3320	5210	91.5	.0216	82.5	.0183	8.6	37.7
16	193	43.3	3070	4220	90.5	.0198	82.3	.0176	8.1	29.2
17	204	64.1	2440	2910	96.7	.0226	90.6	.0209	9.4	35.2
18	204	64.3	2570	2950	97.4	.0229	91.0	.0209	9.8	28.4
19	206	43.0	2940	2680	107.0	.0141	80.1	.0131	8.2	26.3
20	213	42.2	3130	1880	87.8	.0135	80.3	.0123	8.5	26.9
21	226	42.5	2900	3440	83.5	.0172	77.0	.0154	8.7	34.5
22	235	42.2	3220	1980	87.2	.0180	80.2	.0119	8.8	31.4
23	245	44.6	3300	3490	88.5	.0164	81.6	.0148	9.2	36.0
24	266	42.5	3130	2350	82.0	.0141	76.0	.0130	9.8	32.4
25	267	45.1	3290	3430	88.3	.0159	82.0	.0147	9.4	35.3
26	268	42.8	3400	2820	84.4	.0146	77.9	.0133	10.1	36.7
27	273	46.9	4220	6400	97.9	.0254	90.0	.0225	10.2	42.2
28	274	75.5	4400	6620	98.6	.0254	90.4	.0227	10.7	38.6
29	275	56.4	3810	5750	100.0	.0257	92.9	.0233	10.8	40.1
30	280	42.9	3090	3550	80.7	.0158	75.1	.0146	10.0	33.4
31	284	73.0	2830	3740	103.1	.0278	97.9	.0262	11.8	44.0
32	285	73.0	2800	3680	103.0	.0278	97.9	.0265	11.5	33.1
33	290	48.3	4690	2830	96.2	.0147	88.0	.0135	11.8	43.3
34	292	66.4	3540	4940	106.6	.0272	100.3	——	10.9	43.1
35	302	55.5	4290	6100	104.3	.0253	97.0	.0227	10.8	44.9
36	356	45.3	4440	4960	89.6	.0187	83.1	.0170	12.2	39.0
37	358	46.2	3480	4460	83.2	.0188	78.2	.0173	11.0	42.3
38	358	47.1	3840	5960	86.5	.0207	80.9	.0187	11.9	46.2
39	358	54.0	4070	6030	93.8	.0230	87.9	.0211	12.2	43.4
40	359	45.3	4150	4140	87.6	.0182	81.6	.0168	12.0	46.8
41	361	63.6	2170	1200	86.3	.0164	83.2	.0160	11.6	35.3
42	363	64.6	2040	1510	86.3	.0167	83.4	.0160	11.2	47.2
43	363	73.2	2730	4600	95.2	.0288	91.3	.0272	12.5	50.2
44	369	85.5	1470	——	101.3	.0282	99.2	.0281	11.2	——
45	370	85.7	1610	——	102.0	.0262	99.7	.0262	11.9	——
46	370	45.2	4050	3550	84.6	.0169	74.0	.0157	12.2	38.0
47	370	47.0	4250	6530	88.1	.0223	82.2	.0204	11.3	40.8
48	370	44.3	4480	6180	88.9	.0233	82.7	.0212	11.9	43.2
49	371	85.4	1540	——	100.3	.0280	98.1	.0257	12.5	——
50	371	85.5	1610	——	100.6	.0265	98.3	.0266	13.1	——
51	374	74.6	2540	4840	101.4	.0300	97.9	.0286	11.5	42.6
52	374	45.6	4350	5550	88.0	.0221	82.0	.0203	12.3	40.2
53	375	74.5	2620	4800	101.5	.0301	97.9	.0286	11.8	46.7
54	376	46.6	4370	4050	89.6	.0176	83.6	.0163	12.4	41.6
55	377	49.1	3890	9320	88.0	.0268	82.8	.0238	11.8	55.7
56	378	47.1	3770	2440	85.0	.0132	80.0	.0124	11.9	42.4
57	378	48.5	4260	6980	88.5	.0232	82.8	.0209	12.5	50.2
58	379	48.1	4620	6020	92.2	.0210	86.0	.0191	12.5	50.2
59	379	48.1	3770	6090	86.0	.0214	81.0	.0195	11.9	50.7
60	381	53.1	4810	7840	103.5	.0272	97.4	.0247	11.4	46.8
61	382	53.3	4600	7380	102.3	.0263	96.1	.0236	11.5	56.0
62	383	63.7	3440	5960	99.3	.0274	94.7	.0251	11.9	56.1
63	385	52.7	4640	7160	101.8	.0270	95.6	.0233	11.3	48.0
64	386	49.4	5490	7260	99.2	.0246	92.0	.0186	12.7	47.7
65	387	48.1	5560	6210	99.9	.0225	92.5	.0207	12.5	41.6
66	389	63.7	3490	5830	98.3	.0265	93.7	.0247	12.2	44.6
67	389	49.6	6250	9080	107.0	.0287	98.7	.0259	12.6	44.3
68	393	46.8	4550	8330	94.0	.0262	88.1	.0236	11.6	49.6
69	406	48.3	4290	6120	93.5	.0278	87.0	.0209	11.6	43.2
70	436	42.2	4230	4190	83.5	.0146	78.6	.0135	12.3	51.9

uring the heat and especially the mass transfer was evident in several individual points of data.

A further presentation of the data in generalized coordinates was not included here since the temperature dependence of the coefficients is small and the evaluation of the coefficient of molecular diffusion and thermal conductivity for the air-vapor mixture is somewhat a matter of conjecture.

CONCLUSIONS

It was the purpose of this paper to present experimental data for the heat and the mass transfer on a bare tube surface coil. The addition of the experimentally determined coil surface temperature provided a direct evaluation of the heat and mass transfer coefficients. This measured surface temperature was found to be much lower than would be expected from the present methods for surface temperature calculations. These differences have been explained by the variation with air velocity of the dimensionless ratio designated the Lewis Relation.

The Lewis Relation has an exponential effect on the air states as the air progresses through the coil. For a small change in the Lewis Relation, a large change of the coil surface temperature is required to produce a given coil performance.

The results presented were tabulated and plotted for the coil face velocity at the given entering air state. This was found to be satisfactory, since the dependence of the coefficients on the temperature was negligible over the data range. Thus the summary equations for the coefficients (Equations 5 and 6) were given as a function of the face velocity. For analysis, these equations give the results in a more useful form than the points of data in Table I. These correlating Equations, 5 and 6, can be used in deriving the Lewis Relation, Equation 8.

The coefficient of heat transfer for dry cooling was found to be lower than when the coil was operated with dehumidification. One would expect that the friction factor was lower also for dry cooling. Unfortunately, the core pressure drop taken with these data was

(*Continued on page* 91)

ASHRAE Staff Moves into New Headquarters

The month of September traditionally has signified a "beginning, a fresh start" . . . a new school term, a return to work with renewed spirit following vacation, a brighter outlook with the onset of cooler weather. For ASHRAE, it means the dawn of a new era.

The ASHRAE staff has moved into new headquarters in the United Engineering Center, 345 East 47th Street, New York 17, N. Y., diagonally across from United Nations Plaza. For the first time since the merger in 1959, the personnel of the downtown and Fifth Avenue offices are united. Staff offices on the fourth floor of this gleaming 20-story monument to engineering progress are spanking clean and still redolent of fresh paint and new furnishings. The accompanying illustration shows the UEC Building as viewed across First Avenue, looking South and West. An additional view, as toward the 47th Street entrance appears on the cover of this issue.

ASHRAE is now one of 19 tenant engineering societies in a building which was little more than an engineer's "dream" several years ago. Not only does this move effect a UNITY in our staff operations, it provides a UNITY in the Engineering Profession. About 650 headquarters personnel are occupying this Center. The societies which they represent have a combined membership of more than 300,000 Engineers. ASHRAE's new neighbors are:

American Institute of Chemical Engineers
American Institute of Consulting Engineers
American Institute of Electrical Engineers
American Institute of Industrial Engineers
American Institute of Mining, Metallurgical and Petroleum Engineers
American Society of Civil Engineers
American Welding Society
Engineering Foundation
Engineering Index
Engineering Societies Library
Engineers' Council for Professional Development
Engineers Joint Council
Illuminating Engineering Society
Society of Women Engineers
The American Society of Mechanical Engineers
The Municipal Engineers of the City of New York
United Engineering Trustees, Inc.
Welding Research Council

The importance of this Center as a symbol of engineering advancement can be attested to by the thousands of volunteer financial contributions towards its construction by industrial corporations and members of the societies. Approximately 1,500 ASHRAE members have pledged more than $48,000 to the United Engineering Center Building Fund. Let us not forget, however, that our total pledge is $250,000. We are a long way from our goal.

Although there is no relationship between our tenancy in the Center and the fulfillment of our voluntarily-pledged quota, there is a moral obligation to contribute the entire amount of our fair share in the building's construction. Seventy-nine of our chapters have organized fund-raising campaigns; nine are still delinquent. Our participation — 1,500 contributions from a membership of 18,500 — is exceedingly poor.

Now that our move is a reality, let us renew our endeavors to honor our responsibilities, both as engineers and as grateful citizens of a country in which money can be raised VOLUNTARILY to further the aims of peace and to enhance the well-being of mankind.

PRESIDENT'S PAGE
ELMER R. QUEER
Guest Sponsor

Engineers Joint Council and ASHRAE

ELMER R. QUEER

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(*Continued on page* 96)

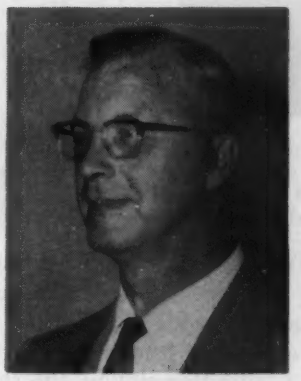
Elmer R. Queer, Presidential Member of ASHRAE, is Director of Engineering Research, Pennsylvania State University.

Degradation of Polyester* Film by Alcohols when used as additives in refrigeration systems

In March 1958 the author's company introduced a new insulation system for use in stators applied to hermetically sealed refrigeration and air conditioning systems. The new insulation system consists of (1) Acrylic coated magnet wire, (2) Polyester film sheet insulation and (3) Polyester covered lead cable and polyester lacing cord. We believe that this insulation system has superior properties, such as higher temperature capabilities, non-water forming and the possibility of shorter, higher temperature dehydration. The properties of this system are described in greater detail by Hurtgen and Mounce.[1]

However, validation activity to determine the full extent of applicability of this new system has shown it to be vulnerable to methanol. The point of alcohol attack is the polyester film sheet insulation, chemically designated polyethylene terphthalate. Methanol at elevated temperatures degrades the film to the point of embrittlement. This has been established by sealed tube work and compressor testing. The preliminary work in sealed tubes gave results which indicated that methanol tends to embrittle the film. These reusable steel tubes are 14 in. long with a 1½-in. diam, have an approximate volume of 350 ml, and are shown in Fig. 1. The over-all length is approximately 17 in. and they can be taken apart readily at the "Ermeto fitting."

Each tube contained 35 gm Suniso 3G oil, ten gm polyester film, ten gm acrylic magnet wire and five gm aluminum casting sprue. Twelve of these tubes containing the above items were dehydrated and evacuated for four hours at 150 C. After cooling, 24 mg of methanol were added to each of six tubes. To three of these, seven gm of Refrigerant 12 were added, and to the other three, five gm of Refrigerant 22. The remaining six tubes contained refrigerant with no methanol; three with seven gm of Refrigerant 12, and three with five gm of Refrigerant 22. Four tubes (one of each variety) could therefore be aged at each of three temperature levels. The condition of the film with and without methanol vapor after 30 days aging at 130, 140 or 150 C is given in Table I.



CLAUS J. BUSHOUSE

Table I

30 Day
Steel Sealed Tube Test

	Tube	Condition of Film
130 C	R-12	flexible
	R-12+MeOH	brittle at ¼ in. r
	R-22	flexible
	R-22+MeOH	brittle at ¼ in. r
140 C	R-12	brittle at ¼ in. r
	R-12+MeOH	brittle at ½ in. r
	R-22	flexible
	R-22+MeOH	brittle at ¼ in. r
150 C	R-12	brittle at ¼ in. r
	R-12+MeOH	brittle at ½ in. r
	R-22	just brittle (at 180° bend)
	R-22+MeOH	shatter brittle

It is evident that the presence of methanol is linked with a more brittle film. Follow-up work using two ¼-hp freezer units also produced quite brittle film. Each unit contained 180 gm Refrigerant 22 plus one cc of methanol. One was run at a winding temperature of 130 C for 82 days and the other at 140 C for 60 days.

Since methanol reportedly is utilized to a limited extent in refrigeration units to prevent capillary freeze-up, it was necessary to investigate this situation and try to disclose some more favorable antifreeze compounds. First of all, it must be stated that the use of any antifreeze is not recommended for the acrylic-polyester insulation system. It is felt that there is no need since this system is non-hygroscopic and thus generates little or no water with little chance for a capillary tube freeze-up.

Investigation into the reactivity of possible antifreezes, therefore, included eight alcohols and two additional solvents, p-dioxane and tetrahydrofuran, both soluble in water and oil. Other antifreeze compounds such as glycols had too high boiling points and could sludge the system; aldehydes are prone to polymerize and ketones may halogenate. The best additive, therefore, had to be a good antifreeze, preferably less reactive than water, and should be compatible with all components of the system.

PROCEDURE

The film was heated in a specially designed tube in the presence of each of these compounds, and also water, whose hydrolytic reactivity was used as a sort of standard. The film also is sensitive to hydrolysis by water. However, this reaction

* Mylar of E. I. duPont de Nemours & Co., Inc.

Claus J. Bushouse is active in the Hermetic Motor Department of the General Electric Company. This paper was presented at the ASHRAE Annual Meeting, Denver, Colo., June 26-28, 1961.

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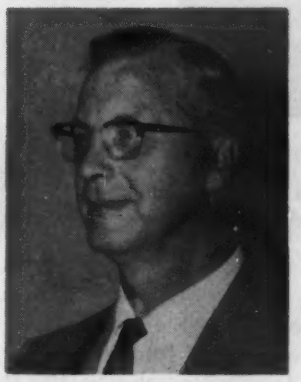
In March 1958 the author's company introduced a new insulation system for use in stators applied to hermetically sealed refrigeration and air conditioning systems. The new insulation system consists of (1) Acrylic coated magnet wire, (2) Polyester film sheet insulation and (3) Polyester covered lead cable and polyester lacing cord. We believe that this insulation system has superior properties, such as higher temperature capabilities, non-water forming and the possibility of shorter, higher temperature dehydration. The properties of this system are described in greater detail by Hurtgen and Mounce.[1]

However, validation activity to determine the full extent of applicability of this new system has shown it to be vulnerable to methanol. The point of alcohol attack is the polyester film sheet insulation, chemically designated polyethylene terphthalate. Methanol at elevated temperatures degrades the film to the point of embrittlement. This has been established by sealed tube work and compressor testing. The preliminary work in sealed tubes gave results which indicated that methanol tends to embrittle the film. These reusable steel tubes are 14 in. long with a 1½-in. diam, have an approximate volume of 350 ml, and are shown in Fig. 1. The over-all length is approximately 17 in. and they can be taken apart readily at the "Ermeto fitting."

Each tube contained 35 gm Suniso 3G oil, ten gm polyester film, ten gm acrylic magnet wire and five gm aluminum casting sprue. Twelve of these tubes containing the above items were dehydrated and evacuated for four hours at 150 C. After cooling, 24 mg of methanol were added to each of six tubes. To three of these, seven gm of Refrigerant 12 were added, and to the other three, five gm of Refrigerant 22. The remaining six tubes contained refrigerant with no methanol; three with seven gm of Refrigerant 12, and three with five gm of Refrigerant 22. Four tubes (one of each variety) could therefore be aged at each of three temperature levels. The condition of the film with and without methanol vapor after 30 days aging at 130, 140 or 150 C is given in Table I.



CLAUS J. BUSHOUSE

Table I

30 Day
Steel Sealed Tube Test

	Tube	Condition of Film
130 C	R-12	flexible
	R-12+MeOH	brittle at ¼ in. r
	R-22	flexible
	R-22+MeOH	brittle at ¼ in. r
140 C	R-12	brittle at ¼ in. r
	R-12+MeOH	brittle at ½ in. r
	R-22	flexible
	R-22+MeOH	brittle at ¼ in. r
150 C	R-12	brittle at ¼ in. r
	R-12+MeOH	brittle at ½ in. r
	R-22	just brittle (at 180° bend)
	R-22+MeOH	shatter brittle

It is evident that the presence of methanol is linked with a more brittle film. Follow-up work using two ¼-hp freezer units also produced quite brittle film. Each unit contained 180 gm Refrigerant 22 plus one cc of methanol. One was run at a winding temperature of 130 C for 82 days and the other at 140 C for 60 days.

Since methanol reportedly is utilized to a limited extent in refrigeration units to prevent capillary freeze-up, it was necessary to investigate this situation and try to disclose some more favorable antifreeze compounds. First of all, it must be stated that the use of any antifreeze is not recommended for the acrylic-polyester insulation system. It is felt that there is no need since this system is non-hygroscopic and thus generates little or no water with little chance for a capillary tube freeze-up.

Investigation into the reactivity of possible antifreezes, therefore, included eight alcohols and two additional solvents, p-dioxane and tetrahydrofuran, both soluble in water and oil. Other antifreeze compounds such as glycols had too high boiling points and could sludge the system; aldehydes are prone to polymerize and ketones may halogenate. The best additive, therefore, had to be a good antifreeze, preferably less reactive than water, and should be compatible with all components of the system.

PROCEDURE

The film was heated in a specially designed tube in the presence of each of these compounds, and also water, whose hydrolytic reactivity was used as a sort of standard. The film also is sensitive to hydrolysis by water. However, this reaction

* Mylar of E. I. duPont de Nemours & Co., Inc.

Claus J. Bushouse is active in the Hermetic Motor Department of the General Electric Company. This paper was presented at the ASHRAE Annual Meeting, Denver, Colo., June 26-28, 1961.

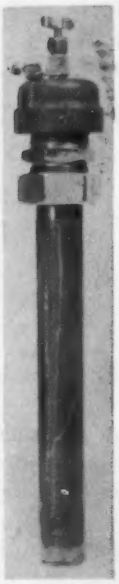


Fig. 1 Reusable steel tube for sealed tube aging

Fig. 2 Glass reaction tube with film and alcohol

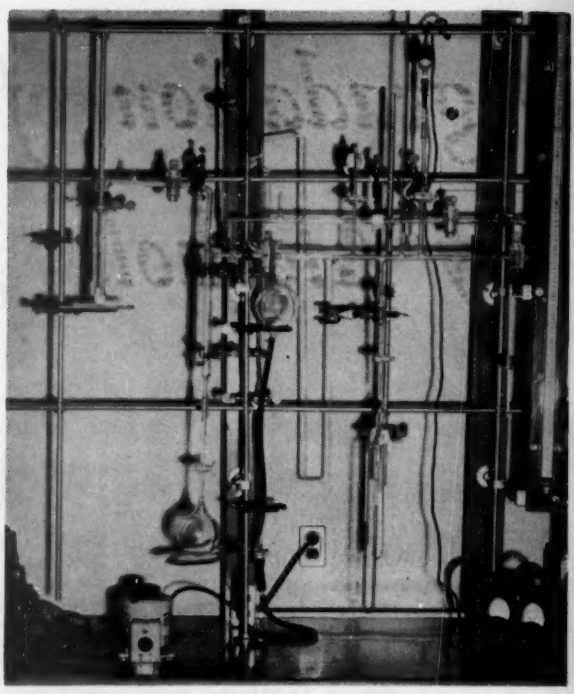


Fig. 3 Glass apparatus for evacuation of reaction tube and addition of alcohol

is quite dependent upon the concentration and the temperature. At room temperature, hydrolysis of the film is measured in centuries, while at 200 C, in fractions of hours.

The glass apparatus designed for this study is pictured in Fig. 2. It is fabricated from 1-in. and 7-mm Pyrex tubing and has an overall length of about 19 in. The self-contained, closed-end mercury manometer is capable of readings up to about 250 mm and the entire apparatus can be placed in a 20-in. oven to follow pressure changes through the glass observation panel. No transpiration corrections are necessary because both manometer and reaction tube are at the same reaction temperature. The alcohol in vapor form is added in sufficient quantity so that at reaction temperature the completely vaporized compound does not exceed the manometer limitations. The volume of the tubes used for the runs was relatively constant at 140 ml ± five ml so that it was necessary to follow pressure changes only.

The partially fabricated tube is cleaned thoroughly with detergent, rinsed with C. P. acetone, and dried at 105 C. After cooling, sufficient triple distilled mercury is added to the tube, which then is warmed slightly and evacuated to less than 20 micron using a clean rubber stopper as a closure. The tube is tilted and mercury is allowed to run into the manometer. Air is admitted slowly, the stopper removed, and 20 gm of ten-ml polyester film strips, 6 x ½ in., are dropped into the tube. The tube is then finished by sealing on a glass cap. The cap is a one-in. piece of pyrex tubing of the same diameter as the body of the reaction tube, to which a small stopcock has been sealed. The upper end of this stopcock has been sealed to a ground joint for attachment to the evacuation system, Fig. 3. The tube is heated with heating tape to 150-160 C and the system and tube are evacuated for six or more hours until the pressure is below ten micron. After cooling down to dry ice-acetone temperature, dry air-free alcohol, as a vapor at room temperature, is condensed into the tube using the evacuation system manometer as a gauge. It is then ready to be sealed off from the system.

Since the usable volume of this specific system is 810 ml, a certain manometer reading difference would indicate the condensation of a known suitable amount of alcohol into the reaction tube. During the condensation of methanol or water into the tube, the pressure decrease was held at 18-20 mm Hg. In the case of methanol this indicated that about 25 mg had been introduced into the tube. The weights of the other alcohols would be in direct proportion of their molecular weights to methanol. The alcohol is contained in a small storage tube (upper left, Fig. 3) attached to the system and is dehydrated in situ (a few hours before use) using calcium oxide and refluxing gently for ten min. This tube is cooled with dry ice and evacuated to remove air; as a further precaution, just before condensation into the reaction tube, the alcohol is allowed to expand into the system four or five times and evacuated each time.

The prepared tube is placed into an oven with an observation panel in the door and a steel millimeter scale is positioned between manometer and tube. Pressure readings are taken easily from the outside using flashlight and reading glass. Reading time intervals vary from five min to twice a day. The higher temperatures obviously require more frequent measurements. The materials used were as follows:

Polyester film (polyethylene terphthalate) 1000A Type A.
Methanol, Baker Analyzed Reagent.
Ethyl alcohol, 190° grain Neutral Spirits (Publicker).
N-propyl alcohol, MCB 2852.
Isopropyl alcohol, Fisher Certified A-416.
N-butyl alcohol, Baker Analyzed Reagent.
Isobutyl alcohol, MCB 2858.
Sec-butyl alcohol, MCB 5047.

Tertiary butyl alcohol, Eastman 820.
Tetrahydrofuran, Baker Analyzed Reagent.
P-dioxane, Eastman 2144.

Although some of the above compounds were known to have properties causing them to be of doubtful use as antifreezes, the experiment was carried out with each one in order to make the study as complete as possible.

RESULTS

Pressure readings in millimeter of mercury were plotted on semi-log paper against linear time in minutes. The log pressure values follow a straight line and the reaction rate could readily be calculated for each temperature by using the following equation:

$$k_T = \frac{2.303}{t_2 - t_1} \times \log \frac{P_1}{P_2}$$

This is the integrated form of $\frac{-dc}{dt} = kc$ with pressure used in place of concentration when volume is constant

Where k_T = specific reaction rate
$t_2 - t_1$ = time interval during which pressure drops from P_1 to P_2.

The specific reaction rates for each reactive compound at three or more temperatures then were plotted on semi-log paper against the reciprocal of the absolute temperature. The position and slope of these lines give an indication of the relative reactivity of the alcohols studied; the smaller the number at a specific temperature, the less reactivity. For example, a rate of 1×10^{-4} or 0.0001 means that 0.01% of the alcohol is reacting with the film each minute. The Arrhenius equation $\frac{d\ln k}{dT} = \frac{E}{RT^2}$ whose integrated form is $\log k = \frac{E}{2.303 \times R}\left(\frac{1}{T}\right) + C$, is the basis for this plotting method, which shows that a straight line results from plotting log k against 1/T.

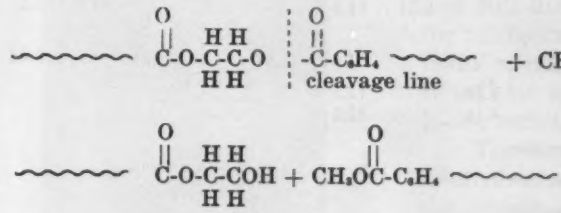
The experimental results are tabulated in Table II, and plotted in Figs. 4, 5 and 6. According to these results, the relative reactivity of the compounds studied is as follows:

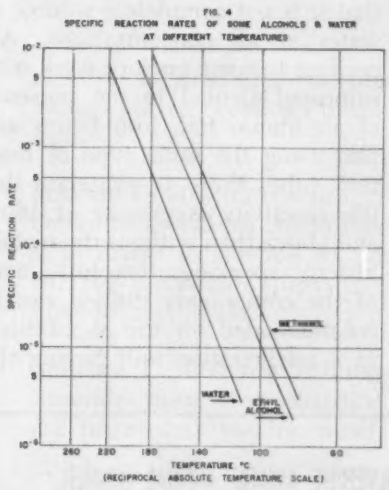
1. No reaction
   p-dioxane
   tetrahydrofuran
   tertiary butyl alcohol
2. Some reactivity
   water
   secondary alcohols—
   isopropyl alcohol
   secondary butyl alcohol
3. Good reactivity
   primary alcohols—
   methanol
   ethyl alcohol
   normal propyl alcohol
   normal butyl alcohol
   isobutyl alcohol

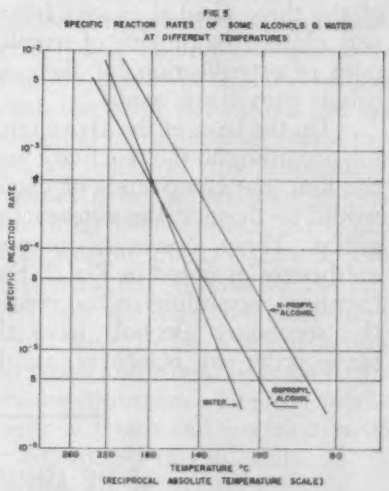
DISCUSSION

Reaction between the alcohols and the film is termed "alcoholysis." In this instance it is a simple ester interchange. Polyethylene terphthalate has a molecular weight of about 15,000-16,000 and is a polymer consisting of about 75-80 units, each having a molecular weight of 192. Embrittlement of the film by an alcohol, methyl, for example, involves the rupture of at least one of the approximately 150 ester linkages in the chemical equation. The chain has now been broken, in the simplest case, into two fragments; one fragment ending in the half ester of ethylene glycol and the other fragment in a methyl ester.

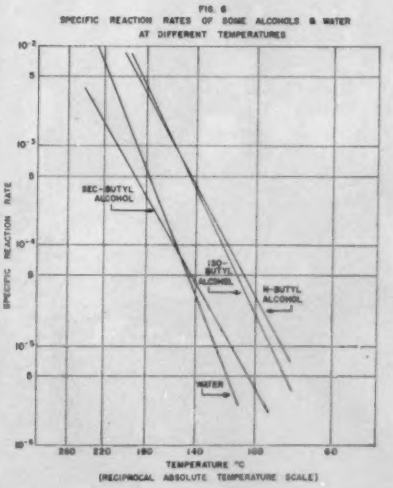
Reactivities of the alcohols with the polyester film fall into groups corresponding to the three alcohol classes; primary, secondary and tertiary. Primary alcohols were found to be the most reactive, ter-







Figs. 4-6 Specific reaction rates per minute for some alcohols and water with polyester film. Fig. 7 Freezing points of aqueous solutions



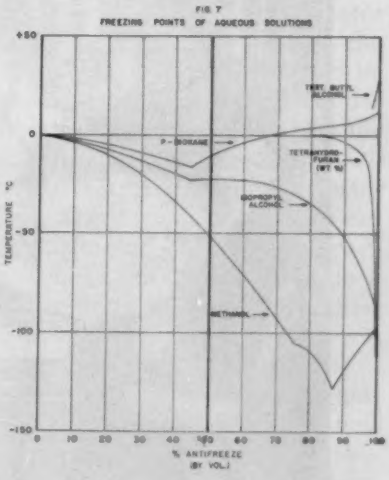


Table II

Specific Reaction Rates With 10 Mil Polyester Film
(Multiply by 10^{-4} min^{-1})

	120 C	130 C	140 C	150 C	160 C	170 C	175 C	180 C	197 C	200 C
Water					1.44			5.29		19.6
					1.48			5.54		21.3
Methanol	1.25	3.43	5.00		23.6					
	1.37		7.40		28.8					
Ethyl alcohol		1.19	3.36	5.45	10.4	20.0				
	(a)	1.97								
N-propyl alcohol .	1.31	1.81	2.91	6.13	10.9			32.4		
			4.26							
Isopropyl alcohol .	0.193		0.535	0.885	1.78	2.72				
			0.623		2.07					
N-butyl alcohol ..				6.31	13.3		25.2		63.7	90.5
							31.2			
Sec-butyl alcohol .					1.48	1.67		3.21		8.44
Isobutyl alcohol ..	1.02		3.78		12.3					
	0.856		4.16		12.8					
Tert-butyl alcohol	No reaction									
Tetrahydrofuran	No reaction									
P-dioxane	No reaction									

Note (a) = 136 C

tiary butyl alcohol had no reactivity and the two secondary alcohols were in between. The reactivities of the three alcohol classes follow very closely the ranking of reaction rates of esterification for these alcohols with acetic acid.[2]

On the basis of the above ranking according to the reactivity with the film, the compounds of choice would be those in the non-reactive group. These, however, are poor antifreezes as noted in Fig. 7. Furthermore, according to the results, the secondary alcohols have the same order of reactivity as the water whose freezing point is to be lowered. Secondary butyl alcohol has another disadvantage in that it is not completely soluble in water at all concentrations. According to some cursory work with isopropyl alcohol in the presence of aluminum foil, iron filings and film using the same type of reaction tube, there is evidence that the reactivity is greater at 160 C and above than without the metals. Strictly speaking, therefore, none of the compounds studied can be recommended for use as additives in a refrigeration unit having this polyester film as sheet insulation in the hermetic motor.

SUMMARY

Since the possibility of the use of methanol as a remedial measure in hermetic refrigeration units exists, various compounds were studied in order to determine their reactivity with polyethylene terphthalate film used in the new insulation system. Preliminary work showed that methanol is quite reactive, causing rapid embrittlement of the film. Studies of each alcohol with the film in a specially designed glass tube indicated that methanol and four other primary alcohols have the same order of reactivity; that two secondary alcohols were about equal to water in reactivity and that tertiary butyl alcohol had no reactivity. No recommendations can be made since the most likely substitute, isopropyl alcohol, is just as reactive as water; it might fall short in antifreeze properties, and it might cause an increase in non-condensable gases when used in the hermetic system. The conclusion, therefore, is that any of the ordinary antifreeze alcohols when added to the system would cause rapid degradation of the polyester film in this new insulation system.

REFERENCES

1. J. P. Hurtgen and A. R. Mounce, "New Insulation System for Hermetic Refrigeration Motors," ASHRAE JOURNAL, p 60, July 1959.
2. G. H. Richter, "Textbook of Organic Chemistry," 2nd. ed., p 191, Wiley, New York, 1945.

AT THE ASHRAE-WOLVERINE AWARD ANNUAL DINNER

Following the custom inaugurated at the previous 1960 Annual Meeting in Vancouver, those Wolverine Diamond Key Award Winners attending the 68th Annual Meeting in Denver and a few special invitees, were guests of the Wolverine Tube Division of Calumet & Hecla, Inc., on Monday, June 26 at the Denver-Hilton.

Most recently admitted to this honored group of award winners of the past was B. P. Morabito of the Carrier Corporation whose "How Higher Cooling Coil Differentials Effect System Economics" appeared in the JOURNAL in August 1960.

The award is in recognition of the best paper JOURNAL — published in the March through February annual interval.

Integrated load technique for

Estimating Annual Energy Use

of central air-conditioning plants

Air conditioning service of the required integrity at the lowest total annual cost – that is our objective. However, when various choices are possible, it is sometimes difficult to decide which system design, granting equally acceptable integrity, will result in the lowest total annual cost. This paper describes how such a problem was approached and the results obtained.

The project involved selection of the refrigeration plant to air condition a 5-million-dollar addition to a large naval hospital. The initial capacity required was 900 ton with provision for future expansion to 2700 ton.

Preliminary studies revealed a sizable capital investment would be required to provide electrical power capacity for an all-electric refrigeration plant. Steam of sufficient capacity to supply the air conditioning was available year-round from the existing central heating plant. Building layout made it possible for steam to be brought into the refrigeration building merely by punching a hole in a wall of the adjacent heating plant. Since the ultimate plant will contain multiple units no stand-by capacity was considered necessary.

Five different schemes were evaluated:

Scheme A: 900-ton hermetic centrifugal compressor; electric motor drive; electric auxiliaries.

Scheme B: 125 psi steam to turbines driving the chilled water and condenser circulating water pumps; turbines exhausting at 12 psi to 900-ton absorption machine; electric motors drive cooling tower fans.

A. EUGENE CONGRESS

Scheme C: 900-ton centrifugal compressor, 125-psi steam turbine driver exhausting to 2.5 in. Hg condenser pressure; electric auxiliaries.

Scheme D: 900-ton lithium bromide absorption machine (12 psi steam); electric auxiliaries.

Scheme E: Split cycles: 125-psi steam turbine driving a 300-ton centrifugal compressor; turbine exhausting at 12 psi to a 600-ton absorption machine; electric auxiliaries.

Two aspects of the problem required special study: (1) determination of the initial capital investment and (2) estimating the annual operating cost. Estimates of procurement and installation costs were not difficult to obtain. But the comparative estimates of operating costs—especially energy costs – presented a more difficult problem. Preliminary estimates had been based on a simple rule-of-thumb solution. It assumed equivalent full-load operating hours for the refrigeration machine and total hours operation for the auxiliaries. It will be seen that this approach is not sufficiently accurate.

The major factor in evaluating relative annual operating costs is the energy charge, excluding for the moment the annual cost to amortize the capital investment. But how is energy to be charged?

DETERMINATION OF UNIT ENERGY COST

In deciding whether to purchase extra power or add equipment to generate power in a Navy-owned plant, the Navy determines the incremental or differential cost – the cost of each of the alternative schemes over and above the cost of existing level of operations. Costs of an existing plant, if they do not vary with the alternative considered, are not charged since they would have to be charged across the board and would have no effect on the outcome. Energy increments of steam and electricity will be considered in this study.

Steam—The principal incremental element of steam, as an energy source, is fuel. Whether a 50,000-lb/hr capacity boiler generates 25,000 lb/hr or 50,000 lb/hr, the difference in cost is determined by the amount of fuel consumed for the extra steam produced, plus the minor charges for additional utilities used in generation of the extra steam. These minor utility costs can be discounted when the stepped-up level of operations results in a compensating increase in boiler efficiency.

Electricity – The incremental unit cost of electric energy is the unit cost of that power block superimposed on existing consumption,

A. Eugene Congress is Head, Air Conditioning and Research Section, Power Generation Branch, Bureau of Yards and Docks, Department of the Navy. This paper was presented at the ASHRAE 68th Annual Meeting, Denver, Colo., June 26-28, 1961.

and not the average annual kwhr cost for the whole customer power-consumption complex. For the plant under consideration, the cost consists of an energy charge in the 600,000 kwhr-per-month block, and a monthly kw demand charge for each month of operation.

ESTIMATING ANNUAL ENERGY AND WATER CONSUMPTION

Steps for determining the quantity of energy, the kwhr or lb of steam consumed during an air-conditioning season, and makeup water for the cooling tower:

STEP 1: Determination of Climatic Data and Operation at Various Percentage Loads – The Weather Bureau of the United States Department of Commerce has compiled 5-year summaries of hourly observations of weather for each of 114 U. S. airway stations.* The number of hourly observations is indicated by 5-deg intervals for each month. Table I was compiled from such data for the Washington, D. C., area. It shows the average number of hours of dry-bulb temperatures for the air-conditioning months April-October in 5-deg intervals ranging from 60 to 104 F and the total number of hours in each of 8 temperature ranges.

Fig. 1 is a graphical presentation of the same data with the curve drawn through the mean of each temperature range. To the ordinate is added a percentage load scale (0 to 100) correlated directly to the 60 to 95 F plus temperature range. Dry-bulb temperature, rather than wet-bulb or other measurement, is assumed as more nearly representative of changes in cooling load. The energy consumption of alternative refrigeration

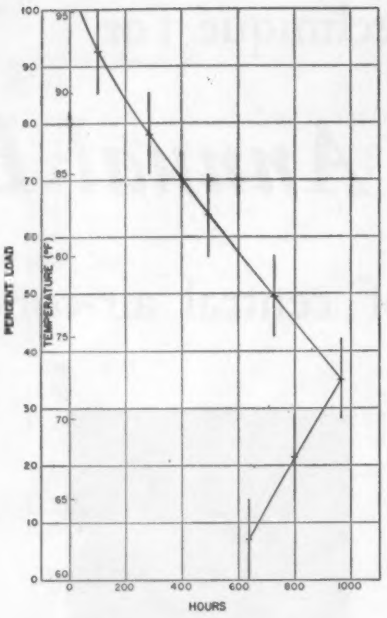


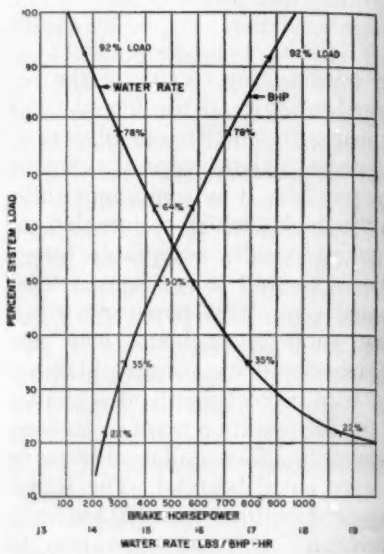
Fig. 1 Temperature and load variation vs. number hours (April-October)

schemes then was evaluated using this base of hours-temperature and percentage load.

STEP 2: Determination of Power Input – This involves the determination of power input to the refrigeration machine at the percentage loads corresponding to the mean temperature of each 5-deg temperature range. Power input of centrifugal compressors and absorption machines as a function of load is supplied by refrigeration equipment manufacturers; water rates (lb/bhp-hr) for steam turbine drives are obtained from turbine manufacturers. Fig. 2 shows the relationship between bhp and water rate to system load for the 970-bhp turbine drive used in the analysis of Scheme C.

STEP 3: Construction of Integrated Load Duration Curves – It is possible to estimate the total energy input for the chiller by multiplying the rate of energy input at various loads by the corresponding load hours. The energy input to auxiliary drives may be added, the amount depending upon the proposed method of operation of these units. However, the "Integrated Load Duration Curve," Fig. 3, is more illuminating and useful in complicated analysis.

Fig. 2 970 hp – 7500 - rpm constant speed turbine. Bhp and water rate vs. system load



The ordinate represents rate of energy input (kw, lb/hr steam, gal fuel oil, cu ft gas or Btu/hr). The abscissa represents the cumulative hours of operation at the various loads. The area under each step (such as ABCD) multiplied by the unit value of one sq in. gives the energy consumption for each partial load – for, kw × hr = kwhr or lb/hr steam × hr = lb steam, etc. Therefore, the total energy area under the steps multiplied by the unit area energy value yields the total energy input. The area under a smooth curve drawn through the midpoint of each riser is equivalent to the area under the steps. Energy consumed by auxiliary drives may be superimposed if of the same type of energy. A planimeter is used to measure the areas. Similarly, a load duration curve may be plotted to obtain ton-hr refrigeration, designating the ordinate as tons.

* ("Climatology of the U. S. No. 30," available from Superintendent of Documents, U. S. Government Printing Office, Washington, D.C.)

Table I—Hours of Temperature Occurrence per Month
(Average of 5 Years: 1950-1955)

Temperature Range F	April	May	June	July	Aug.	Sept.	Oct.	Total Per Temp. Range
104-100			0.4	0.8				32
99- 95			7.2	15.0	4.6	4.4		
94- 90		0.8	27.4	44.8	16.0	10.0	1.6	101
89- 85	2.6	11.6	65.8	105.0	64.4	24.8	9.0	283
84- 80	13.6	34.2	96.5	159.0	120.0	54.8	16.2	494
79- 75	26.2	64.6	135.0	173.0	185.0	105.5	38.0	727
74- 70	42.2	109.0	165.0	166.5	223.0	159.5	98.5	964
69- 65	69.5	151.0	136.5	65.4	97.2	147.0	133.0	800
64- 60	104.0	155.0	58.8	12.2	28.3	121.0	159.0	638

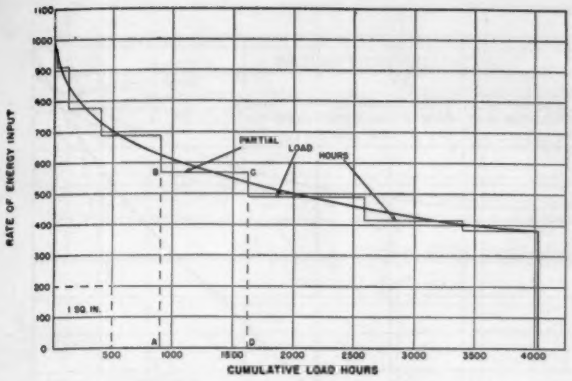


Fig. 3 Typical load duration curve

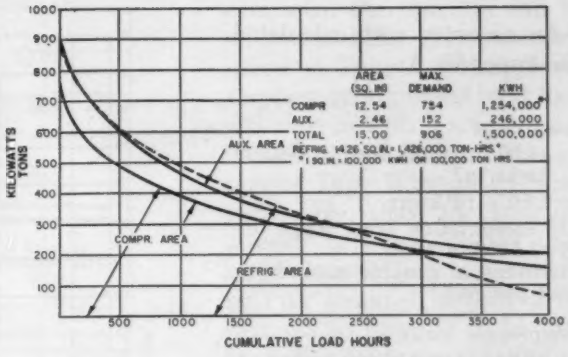


Fig. 4 Scheme A load duration curve (all-electric drive)

INTEGRATED ENERGY AND INCREMENTAL COST PROCEDURES

Application of the integrated energy procedure to 2 of the 5 central-plant alternative designs will be presented herein, together with results obtained from the analysis of all 5 schemes. The data of Table II formed the base for the development of calculations involving power input.

Table II—Per Cent Load at Mean Temperatures and Hours at Part Load Operation

(Developed from Table I and Fig. 1)

Temp. Range F	Per Cent Load Correlated to Mean of Temp. Range	Hrs
95+	100	32
94-90	92	101
89-85	78	283
84-80	64	494
79-75	50	727
74-70	35	964
69-65	22	800
64-60	7	638

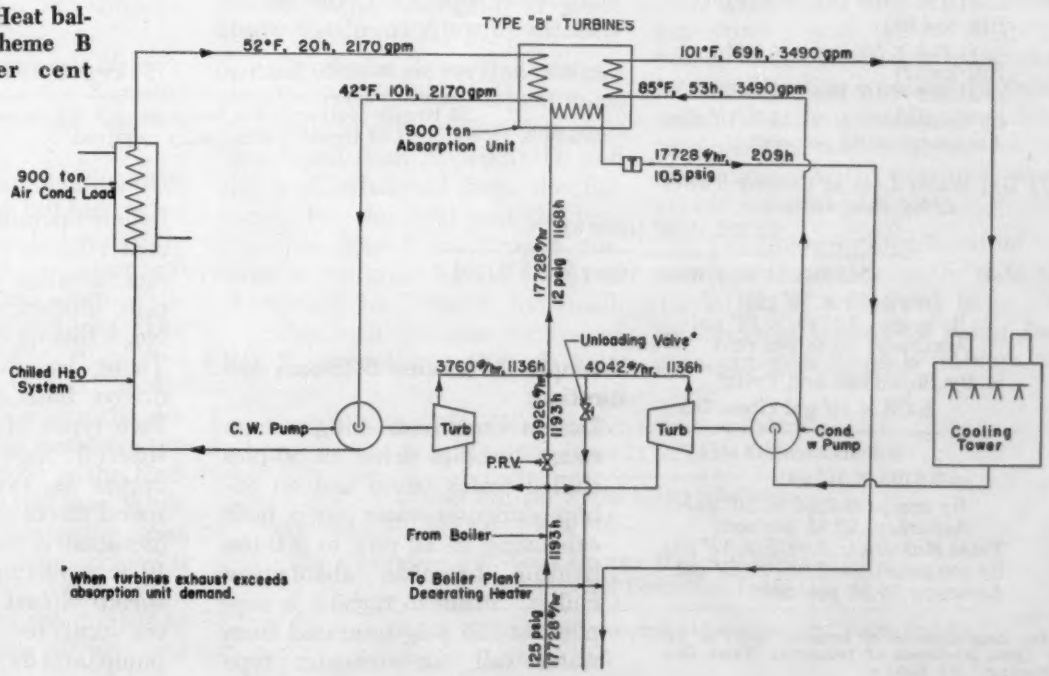
Example I: Scheme A (All-Electric)

Design Conditions: A 970-bhp motor drives a 900-ton centrifugal compressor. Leaving chilled-water temperature: 42 F. Condenser-water inlet temperature: 85 F. Fouling factor: 0.001. At 100 per cent system load the chilled-water pump takes 80 bhp; the condenser-water pump, 62 bhp. Motors on both pumps are two-speed — ½ speed on the chilled-water pump at 50 per cent, and below, system load; 2/3 speed on the condenser-water pump at 64 per cent, and below, system load. Two 20-hp motors drive cooling tower fans —but one on at 50 per cent load and below.

The annual power consumption and water requirement were calculated in the conventional manner and are recorded in Appendix A. From these data the level of power input for compressor and auxiliaries, and ton refrigeration were used to plot the load duration curves of Fig. 4. The area under the lower full-line curve represents annual kwh consumption of the compressor motor; the area between the two full-line curves represents the annual power consumption of the motor-driven auxiliaries. Together these two areas indicate the total annual electrical consumption. The area under the dash-line curve represents the annual ton-hr refrigeration, which is a constant for each of the alternative schemes considered.

From this graph the following information is derived and com-

Fig. 5 Heat balance Scheme B (100 per cent load)



pared for accuracy with calculated data in Appendix A:

(1) Total kwhr
$= 15.00$ in.^{2} total area $\times$ 100,000 kwhr/in.^{2}
$= 1.50 \times 10^6$ kwhr
By computation from Appendix A: 1,505,113 kwhr
Accuracy of graphic method: 99.7 per cent

(2) Compressor kwhr
$= 12.54$ in.$^2 \times 100{,}000$ kwhr/in.^{2}
$= 1.254 \times 10^6$ kwhr
By computation: 1,247,475 kwhr
Accuracy of graphic method: 99.5 per cent

(3) Equivalent Full-Load Energy Hours (Chiller)

$$= \frac{12.54 \text{ in.}^2 \text{ Compr. area}}{3.77 \text{ in. ordinate}} \times 500 \text{ hr/in.}$$

$= 1663$ hr
By computation: 1655 hr
Accuracy: 99.52 per cent

(4) Refrigeration Ton-Hr
$= 14.26$ in.^{2} Refrig. area $\times$ 100,000 ton-hr/in.^{2}
$= 1.426 \times 10^6$ ton-hr
By computation: 1,425,042 ton-hr
Accuracy: 99.93 per cent

(5) Equivalent Full-Load Refrig. Hours

$$= \frac{1.426 \times 10^6 \text{ ton-hr}}{900 \text{ ton}}$$

$= 1584.4$ hr
By computation: 1583 hr
Accuracy: 99.91 per cent
(Note. The area under the curve can be equated to a rectangle using the length of the vertical ordinate as one side.)

(6) Total Btu to Be Removed by Cooling Tower
= System air conditioning load + Compr. motor input
$= (1.426 \times 10^6$ ton-hr $\times 12{,}000$ Btu/ton-hr) $+ (1.254 \times 10^6$ kwhr $\times$ 3,413 Btu/kwhr)
$= 21.392 \times 10^9$ Btu
By computation: 21.36×10^9 Btu
Accuracy: 99.85 per cent

(7) Gal Water Loss at Cooling Tower
a. By Evaporation

$$= \frac{21.392 \times 10^9 \text{ (item 6)}}{1040 \text{ Btu/lb 95 F water vapor} \times 8.3 \text{ lb/gal}}$$

$= 2.478 \times 10^6$ gal
By comp: 2.4745×10^6 gal
Accuracy: 99.88 per cent

b. By Blowdown and Drift*

$$= \frac{2.478 \times 10^6 \text{ gal (item 7a)}}{5 \text{ concentration} - 1}$$

$= 0.6195 \times 10^6$ gal
By comp: 0.6186×10^6 gal
Accuracy: 99.85 per cent
Total Makeup $= 3.0975 \times 10^6$ gal
By computation: 3.093×10^6 gal
Accuracy: 99.86 per cent

* For determination of formula, see Fig. 151 in "Betz Handbook of Industrial Water Conditioning," 5th Edition.

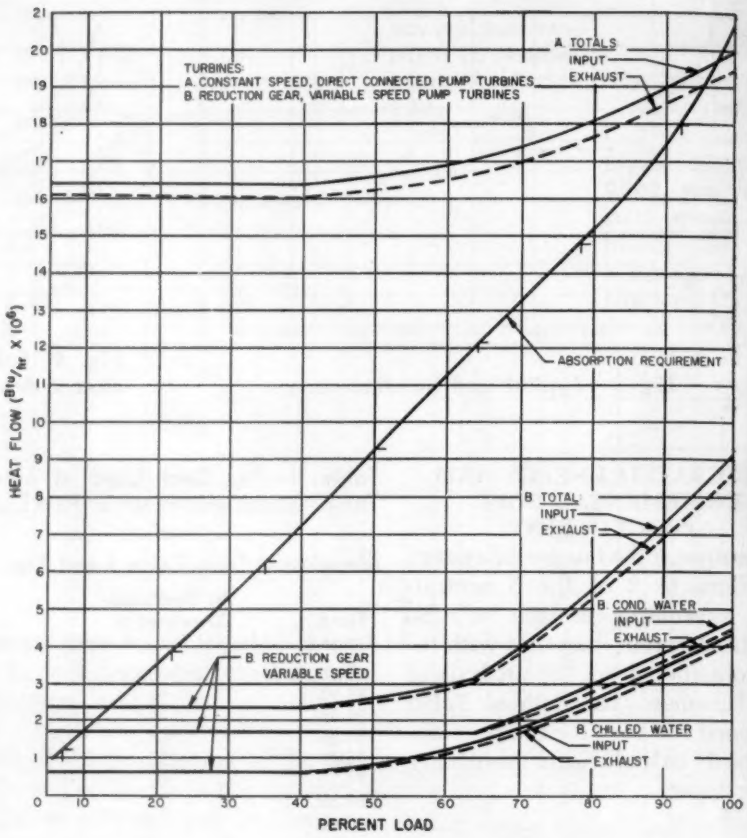


Fig. 6 Scheme B. Absorption and turbine heat flow vs. system load

(8) Gal Condenser Water Required to Remove Heat

$$= \frac{21.392 \times 10^9 \text{ Btu (item 6)}}{10 \text{ Btu/lb cooling tower temp. drop} \times 8.3 \text{ lb/gal}}$$

$= 257.73 \times 10^6$ gal

If condenser water were circulated full flow at all times during compressor operation, the annual quantity of water circulated would

$$= \frac{4039 \text{ hr } [(900 \text{ ton} \times 12000 \text{ Btu/ton}) + (754 \text{ kw} \times 3413 \text{ Btu/kwhr})]}{10 \text{ Btu/lb cooling tower temp. drop} \times 8.3 \text{ lb/gal}}$$

$= 650 \times 10^6$ gal, or 2.52 times that actually required.

Example II: Scheme B (Steam Absorption)

Design Conditions: Single-stage steam turbines drive an 80-bhp chilled-water pump and an 86-bhp condenser-water pump, both exhausting at 12 psig to 900-ton lithium bromide absorption chiller. Steam to turbine is supplied at 125 psig saturated from water-wall, air-preheater type boilers operating at 78 to 82 per cent efficiency. Estimated summer average efficiency 80 per cent. Boilers are fired with F.S. No. 6 fuel oil of 152,200 Btu/gal. There are three 30-hp motor driven fans on cooling tower. Two types of turbines are considered: one (designated on curves as Type A) is constant speed, direct connected; the other (designated on curves as Type B) is reduction geared, variable speed. Speed ranges 40 to 100 per cent for the chilled-water pump, and 64 to 100 per cent for

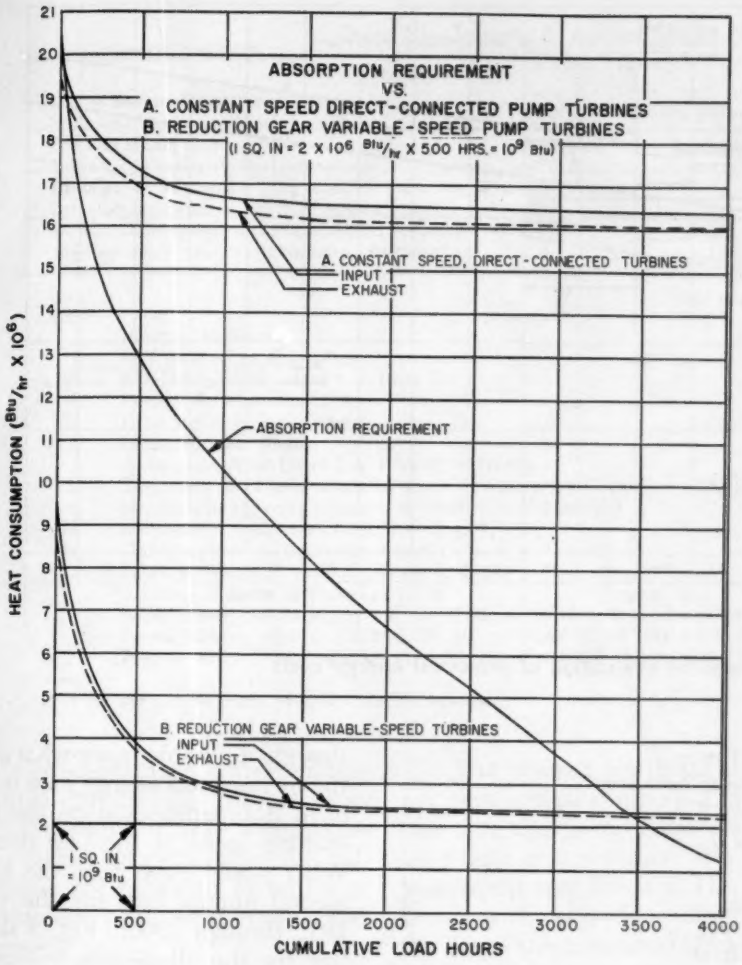


Fig. 7 Load duration curves for Scheme B

the condenser-water pump. Speed of the chilled-water turbine is controlled by a differential pressure regulator in the coil system, and speed of the condenser-water turbine is controlled by temperature of leaving condenser water.

The number of hours of operation is the same at the 8 partial system loads as in the previous all-electric example.

Fig. 5 shows the diagrammatic and heat balance at full load when utilizing high-speed, reduction-gear type turbines. The steam suffers a small thermal drop while passing through the turbines. This drop is shown in Fig. 6 (plotted from heat rates developed in Appendices B and C) where the difference in heat flow between turbine inlet and exhaust is the thermal drop across the turbines. While the turbine drop in passing from maximum load to minimum load is fairly constant for Type A turbines, for Type B turbines the drop is so small at loads below 40 per cent that it cannot be indicated clearly on the scale of this graph. This reduction in turbine drop stems from the fact that the thermal loss across a turbine is inversely proportional to its water rate—lb/bhp hr, the latter increasing at lower loads (see Appendix B).

Fig. 6 also shows the relationship between the rate of heat input required by the absorption unit (developed from Appendix D) and the heat exhausted from the turbines. For the most part the heat flow per hour from Type A turbines is too great for the needs of absorption; for Type B, too small.

The load duration curves of Fig. 7 (plotted from Fig. 6) indicate what the situation will be on an annual basis. The considerable excess of Type A exhaust over absorption requirement will be unwelcome in a plant where it cannot be utilized profitably. The condition of Type B exhaust is less objectionable, since the area of deficiency between exhaust and absorption requirement is made up by PRV steam. The area of excess steam is so small as to be utilized for boiler feedwater heating.

Note that the area on the graph between turbine input and exhaust, representing the turbine thermal drop, is the amount of energy necessary to drive the chilled-water and condenser-water pumps the year round. In the case of the Type B geared, variable speed turbine, it is so small that if disregarded it would be well within estimating accuracy. Only where the final evaluations of alternative choices show a "photo finish" should one be concerned with the thermal drop across auxiliary turbine drives.

The heat balance of Fig. 7, recorded in Table III, shows the marked superiority of a high speed, reduction gear, variable rpm turbine over the constant speed, direct-connected turbine. The latter takes approximately 31.5 million lb boiler steam more than the reduction-gear, variable speed job, and poses the problem of what to do with an almost equal quantity of surplus exhaust. When, as in this case, the by-product steam cannot be utilized profitably, use of the more expensive geared turbine is justified since the initial extra cost can be amortized in a relatively short time. Also, the variable speed feature permits rendering service in accordance with demand, and avoids wasteful pumping.

The following significant information is derived from the data of Table III on an annual basis for refrigeration Scheme B using the reduction-gear Type B turbine:

(1) Fuel Oil Chargeable to Refrigeration System

$$= \frac{24.58 \times 10^9 \text{ Btu (item 11 of Table III)}}{152{,}200 \text{ Btu/gal} \times 0.80 \text{ boiler efficiency}}$$

$$= 201{,}873 \text{ gal} = 4806 \text{ bbl}$$

(2) Equivalent Full-Load Energy Hours (Chiller)

$$= \frac{29.38 \times 10^9 \text{ (item 7 of Table III)}}{20.8 \times 10^6 \text{ Btu (100 per cent absorption heat from Fig. 6)}}$$

$$= 1418 \text{ hr}$$

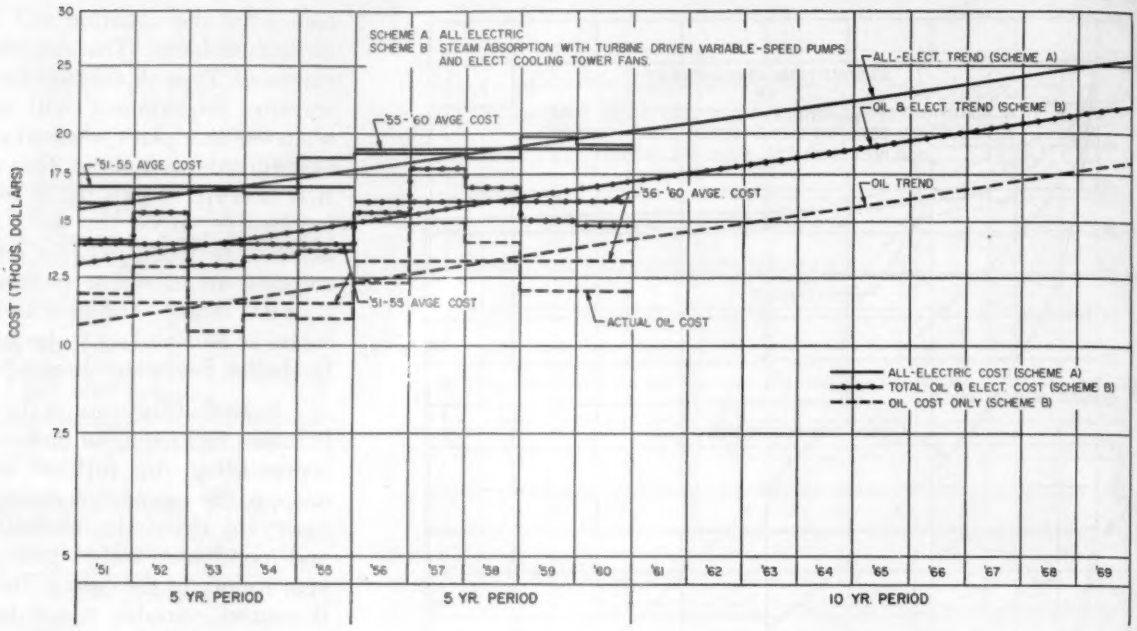


Fig. 8 Comparative evaluation of projected energy costs

(3) Total Heat to Be Removed by Cooling Tower
$= (\text{ton-hr} \times \text{Btu/ton-hr}) + \text{chiller steam-heat drop}$
$= (1.426 \times 10^6 \text{ ton-hr}^*)\ (12000 \text{ Btu/ton-hr})$
$+ 24.10 \times 10^9$ Btu (item 10 of Table III)
$= (17.112 + 24.10)\ 10^9 = 41.212 \times 10^9$ Btu
* (from Example I, item 4)

(4) Gal Water Loss at Cooling Tower
a. By Evaporation

$$= \frac{41.212 \times 10^9 \text{ Btu total heat}}{(1036 \text{ Btu/lb for 101 F water})\ (8.3 \text{ lb/gal})}$$

$= 4.79 \times 10^6$ gal
b. By Blowdown and Drift

$$= \frac{4.79 \times 10^6 \text{ gal}}{5 \text{ concentration} - 1}$$

$= 1.20 \times 10^6$ gal
Total Makeup $= (4.79 + 1.20)\ 10^6$
$= 5.99 \times 10^6$ gal

(5) Condenser Water Required to Remove Heat

$$= \frac{41.212 \times 10^9 \text{ Btu removed by tower}}{(16 \text{ Btu/lb drop} \times 8.3 \text{ lb/gal})}$$

$= 310 \times 10^6$ gal

If condenser water were circulated full-flow at all times during absorption operation, the quantity circulated would

$= 4039$ hr $\times$ 3490 gpm (from Fig. 5) $\times$ 60
$= 846 \times 10^6$ gal
or 2.7 times that actually required.

DETERMINATION OF INCREMENTAL ENERGY COST (ANNUAL BASIS)

As mentioned earlier, in determining the cost of operating a new facility, the Navy employs the incremental rate principle – the cost of that block of new power over and above the existing usage. In the competition between producers of electricity and other types of energy supply, the Navy is neutral. Its prime interest is to discharge its responsibility to taxpayers by using in each application the most economically available energy—all factors considered.

If current unit prices of electricity and fuel oil are used, criticism might be leveled by one supplier or another if its price happened to be momentarily high unless supported by price trend information. It was therefore thought advisable to see what effect the incremental energy load would have if superimposed on the then existing loads of the past decade. What would have been the incremental annual cost for the years 1951 through 1960? Fig. 8 shows this for the all-electric Scheme A and the absorption Scheme B, based on actual unit incremental cost to the government.

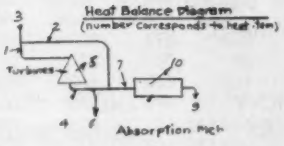
Although the price of oil during that decade fluctuated more than that of electricity (due to the Korean War and the Suez crisis), the graph shows that the trend in energy costs of the two schemes moved side by side. If it can be assumed that this trend will continue in the future, the relative difference in the energy operating costs of the 5 schemes, listed in Table IV and shown graphically in Fig. 9, will also continue.

CAPITAL VALUE OF ANNUAL SAVINGS

While the comparative annual operating costs of Table IV show an annual saving of $5100 for the lowest cost scheme when compared with the highest, this does not by itself justify the installation of the lowest operating cost design. The installed cost of one scheme may be so much less than another that the difference in annual amortized cost would more than offset the savings. An annual savings is equivalent to an annual annuity and can

Table III—Scheme B: Annual Steam Heat Balance (From Fig. 7)

Design Conditions	
Pressure steam to turbine	125 psig
Pressure steam to absorption	12 psig
Enthalpy 125 psig steam (saturated)	1193 Btu/lb
Enthalpy 12 psig steam (saturated)	1162 Btu/lb
Enthalpy saturated liquid from absorption unit	209 Btu/lb
A—Constant-speed, direct-connected turbines	
B—Reduction gear, variable-speed turbines	



Heat Item	Heat	CURVE (See Fig. 7) A	B
		Area in sq. in. or 10^9 Btu (except as noted)	
1	Input to turbines	67.70	11.86
2	PRV makeup to absorption	0.02	18.30
3	Total from boiler (Item 1 + Item 2)	67.72	30.16
4	Exhaust from turbine	66.25	11.38
5	Drop across turbine (Item 1 — Item 4)	1.45	0.48
6	Excess turbine exhaust	36.89	0.30
7	Absorption input (Item 2 + Item 4) — Item 6	29.38	29.38
8	Equivalent lb 12 psig saturated steam to absorption (Item 7 ÷ 1162 Btu/lb)	25.28 ×	10^6 lb
9	In absorption leaving condensate (Item 8 × 209 Btu/lb)	5.28	5.28
10	Drop across absorption (Item 7 — Item 9)	24.10	24.10
11	Chargeable to refrigeration system (Item 5 + Item 10)	25.55*	24.58*
12	Balance (Item 3 = sum of Items 5, 6, 9, 10)		

Excess of curve "A" over curve "B":
Boiler steam (67.72-30.16) 10^9 = 37.56 × 10^9 Btu = 31.5 million lb 125 psig saturated steam
Excess turbine exhaust (36.89-0.30) 10^9 = 36.59 × 10^9 Btu = 31.5 million lb 12 psig saturated steam
Turbine drop (1.45-0.48) 10^9 = 0.97 × 10^9 = 813 thousand lb 125 psig saturated steam

* If full use can be made of excess turbine exhaust

be converted to its corresponding equivalent capital investment, called present worth. The necessary factors for performing this calculation are provided in tables of investment, compound interest and annuity.

The present worth factor at 4 per cent annual compound interest (the rate the government pays for long-term money today) over a 15-year equipment life is 11.118. (This factor will vary, depending upon the life of the amortized equipment and the interest rate the owner must pay for his money.) The present worth calculation then is:

$5,100 × 11.118 = $56,700

This means that to be competitive the total installed first cost of the turbine-driven centrifugal machine

Table IV—Comparative Annual Energy and Water Operating Cost

Design Conditions: 42 F leaving chilled water, 85 F inlet condenser water, 0.001 fouling factor, 4039 total operating hr		Scheme A: All Electric Refrigerant System	B: Steam Pump Drivers Exhaust to Absorption Chiller Elec. Fans Type B Turbine	C: Steam Turbine Driven Refrigerant Compr. Elec. Aux.	D: Straight Absorption Chiller Elec. Aux.	E: Steam Turbine Driven Compr. Exhausts to Absorption Chiller
Quantities						
Electricity	Units					
Energy	kwhr	1,505,113	249,231	430,611	452,877	332,021
Demand						
5 month at	kw	906	99	242	236	178
2 month at	kw	714	99	238	232	174
Fuel oil (F.S. No. 6)	bbl		4,806	5,624	4,806	4,840
Water makeup	10^3 gal	3,093	5,960	6,640	5,960	6,085
Utility Unit Rates (1960)						
Electricity		$	$	$	$	$
Energy	kwhr	0.00731	0.00731	0.00731	0.00731	0.00731
Demand	kw	1.40	1.40	1.40	1.40	1.40
Fuel oil (F.S. No. 6)	bbl		2.516	2.516	2.516	2.516
Water	10^3 gal	0.41	0.41	0.41	0.41	0.41
Water treatment	10^3 gal	0.03	0.03	0.03	0.03	0.03
Costs (1960)						
Electricity						
Energy		$11,002	$ 1,822	$ 3,148	$ 3,311	$ 2,427
Demand (5 month)		6,342	693	1,694	1,652	1,246
Demand (2 month)		2,000	277	666	650	487
Total Electricity		19,344	2,792	5,508	5,613	4,160
Fuel			12,092	14,150	12,092	12,177
Electricity and fuel		19,344	14,884	19,658	17,705	16,337
Water makeup		1,268	2,444	2,722	2,444	2,495
Water treatment		93	179	199	179	183
Total water		1,361	2,523	2,921	2,623	2,678
Total annual energy and water cost		$20,705	$17,507	$22,579	$20,328	$19,015

would have to be $56,700 less than that of equipment of Scheme B. And the motor-driven centrifugal, to be competitive, would have to be $35,600 less than Scheme B.

USE OF EQUIVALENT FULL-LOAD HOURS

This study has revealed that results are likely to be misleading when the same number of equivalent full-load hours is applied to estimate annual energy consumption of alternative types of chillers for the same refrigeration job.

Two different concepts are involved: (1) the output of the plant measured in equivalent full-load refrigeration hr; and (2) the input measured in equivalent full-load energy hr.

(1) Equivalent full-load refrigeration hr

$$= \frac{\text{summation of ton-hr at the various loads}}{\text{refrigeration tons at 100 per cent load}}$$

and is a constant for the same refrigeration job regardless of the type of chiller equipment and kind of energy used. It is the number of hours the chiller operates continuously at 100 per cent load to produce the annual refrigeration effect.

Equivalent full-load energy hours, on the other hand, is not necessarily a constant for alternative chiller equipment considered for an installation.

(2) Equivalent full-load energy hr

$$= \frac{\text{summation of energy consumed at various loads}}{\text{rate of energy input at 100 per cent load}}$$

and is the number of hours the chiller would operate continuously at 100 per cent system power supply to equal its annual energy consumption.

In the present study equivalent full-load refrigeration hr came to 1583 hr – constant for all of the 5 schemes considered. Equivalent full-load energy hr, developed from duration curves (or computation when feasible), is different for different chiller equipment as shown by Table V. If, for example, the 2089 equivalent full-load energy hr of Scheme C were used to estimate annual energy consumption of the other schemes, would not an injustice be done? To take another example: if the 1418 hr of the absorption chiller (Schemes C and D) were applied to the peak compressor kw demand to estimate annual energy consumption of the motor-driven centrifugal, the result would be low by 17 per cent.

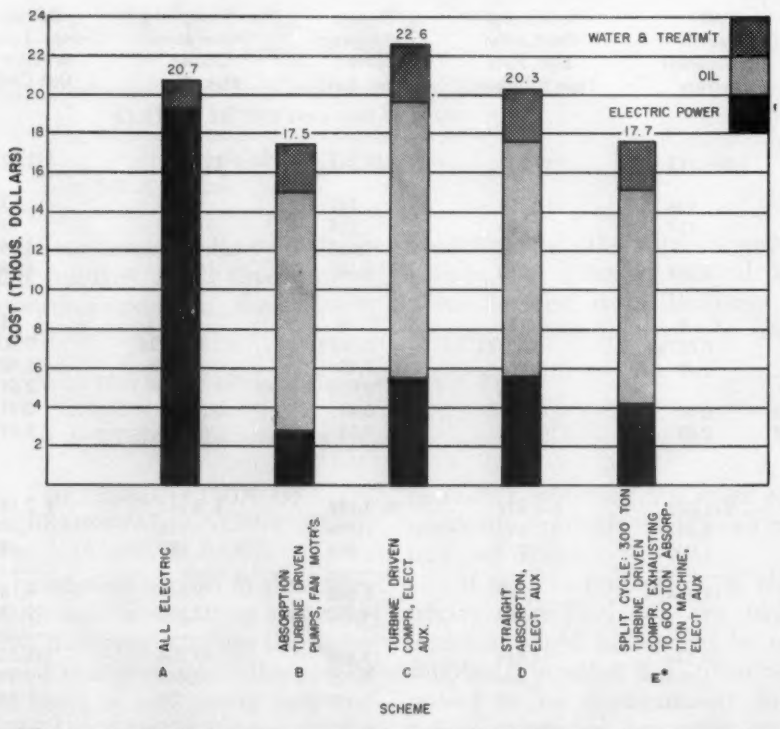
Also, the equivalent full-load energy hours derived by analysis for one job may not be interchangeable with that on another – even using the same type of equipment. Examination of Equation (2) shows that should the chiller operate at a higher or lower system load for a different number of hours—causing a change in the quantity of energy consumption—and thus increase or decrease the numerator, the resulting equivalent full-load energy hours could change. Therefore, one cannot use equivalent full-load energy hours as a short-cut to estimate annual energy consumption unless two conditions are satisfied: (a) the chiller is of the same type and (b) the conditions of operation are identical.

INCREMENTAL BOILER EFFICIENCY

What is a reasonable and justifiable boiler efficiency to use when estimating fuel required to furnish steam for a steam-actuated refrigeration machine? The boilers in the hospital for which this study was made are of the air-preheater water-wall type. Efficiency of operation at half load is 78 per cent, and at full load, 82 per cent. Adding a new summer air conditioning load permits operation at higher than existing summer loads. What will be the efficiency of the added load?

Consider a boiler generating 50,000,000 Btu/hr. The input at 78 per cent efficiency is 50,000,000 Btu/hr ÷ 0.78 = 64,100,000 Btu/hr. If the load on this boiler is increased to 100,000,000 Btu/hr at

Fig. 9 Comparative annual electric, oil and water cost



* To correspond with later revisions to Scheme E of Table IV, the oil portion of the Scheme E column should be raised to a level of approximately 16.3 and the top of the water portion to 19.0.

Table V—Equivalent Full-Load Chiller Energy Hours

Scheme		Hr	Percent in Excess of Scheme B
A	All-Electric	1655	17
B	Steam Turbine-Driven Pumps Exhaust to Absorption	1418	0
C	Steam Turbine Driven Compressor	2089	47
D	Straight Absorption	1418	0
E	Split Cycle	1887	33

Appendix A: Electric Power and Water Calculations for Scheme A

	Column→	1	2	3	4	5	6	7	8	9
		Per Cent System Load								
Compressor (970 bhp)		100 per cent	92 per cent	78 per cent	64 per cent	50 per cent	35 per cent	22 per cent	7 per cent	Total
1. Refrigeration	Ton	900	828	702	576	450	315	198	63	
2. Compr. input as per cent full load	Per Cent	100	90	75	60	48	35	26	20	
3. Compr. input (line 2 × line 3 col 1)	kw	754	679	566	453	362	264	196	151	
4. Hours of operation	Hr	32	101	283	494	727	964	800	638	4,039
5. Electrical consumption (line 3 × line 4)	Kwhr	24,128	68,579	160,178	223,782	263,174	254,496	156,800	96,338	1,247,475
6. Ton-hr (line 1 × line 4)	Ton-Hr	28,800	83,628	198,666	284,544	327,150	303,660	158,400	40,194	1,425,042
7. Equiv. full load energy hour (line 5 col 9 ÷ line 3 col 1)	Hr									1,655
8. Equiv. full load refrig. hour (line 6 col 9 ÷ line 1 col 1)	Hr									1,583
Auxiliaries										
9. Chilled water pump (80 BHP)[1]	kw	67	66	63	60	10	10	10	10	
10. Condenser water pump (62 BHP)[2]	kw	52	52	52	18	18	18	18	18	
11. Cooling tower fans (2-20 HP)[3]	kw	33	33	33	33	17	17	17	17	
12. Total aux. elec. demand	kw	152	151	148	111	45	45	45	45	
13. Total aux. elec. consumption (line 4 × line 12)	kwhr	4,864	15,251	41,884	54,834	32,715	43,380	36,000	28,710	257,638
Compressor and Auxiliaries										
14. Total compr. and aux. demand (line 3 + line 12)	kw	906	830	714	564	407	309	241	196	
15. Total elec. consumption (line 5 col 9 + line 13 col 9)	kwhr									1,505,113
Cooling Tower Duty (Annual Basis)										
Btu to be removed										
16. a. Air conditioning load (line 6 col 9 × 12000 Btu/ton hr)										
								10⁹ Btu	17.10	
17. b. Power input (line 5 col 9 × 3413 Btu/kwhr)								10⁹ Btu	4.26	
18. Total (Sum line 16 and 17)								10⁹ Btu	21.36	
Gal water makeup										
19. a. Loss by evaporation (line 18 ÷ [(1040 Btu/lb for 95 F vapor) × 8.3 lb/gal)]								10⁶ Gal	2.4745	
20. b. Loss by blow down & drift (line 19 ÷ (5 cycles chemical conc. — 1)								10⁶ Gal	0.6186	
21. Total (sum lines 19 and 20)								10⁶ Gal	3.0931	

[1] Two speed motor, ½ speed ≤ 50 per cent load
[2] Two speed motor, ⅔ speed ≤ 64 per cent load
[3] One fan on ≤ 50 per cent load

Appendix C: Pump Turbine Thermal Calculations for Type "A"*

Per cent System Load	BHP	Water Rate lb/bhp hr	Steam Rate lb/hr (Col 2 × Col 3)	Heat Input Btu/hr (1193 Btu/lb × Col 4)	Turbine Heat Drop Btu/lb 2545 Btu/bhp hr / 0.95 × Col 3	Btu/hr (Col 4 × Col 6)	Heat in Exhaust Btu/hr (Col 5 — Col 7)
1	2	3	4	5	6	7	8
			80 hp chilled water pump turbine				
100	80	94	7,520	8,971,360	28.5	214,320	8,757,040
92	79	94	7,426	8,859,218	28.5	211,641	8,647,577
78	75	94	7,050	8,410,650	28.5	200,925	8,209,725
64	70	94	6,580	7,849,940	28.5	187,530	7,662,410
50	66	94	6,204	7,401,372	28.5	176,814	7,224,558
40	63.5	94	5,969	7,121,017	28.5	170,116	6,950,901
			86 hp condenser water pump turbine				
100	86	108	9,288	11,080,584	25	232,200	10,848,384
92	81	108	8,748	10,436,364	25	218,700	10,217,664
78	74	108	7,992	9,534,456	25	199,800	9,334,656
64	72	108	7,776	9,276,768	25	194,400	9,082,368

* Constant speed, direct-connected turbines

Appendix D: Absorption Heat Input*
Steam at 12 PSIG Saturated (1162 Btu/lb)

Per cent System Load	Refrig. Ton	Hours Operation	Steam Rate lb/ton hr	Steam Rate lb/hr (Col. 2 × Col. 4)	Heat Rate Btu/hr (Col. 5 × 1162 Btu/lb)	Total Heat Input 10⁶ Btu (Col. 3 × Col. 6)
1	2	3	4	5	6	7
100	900	32	19.8	17,820	20,706,840	662.62
92	828	101	18.8	15,566	18,087,692	1,826.86
78	702	283	18.5	12,987	15,090,894	4,270.72
64	576	494	18.2	10,483	12,181,246	6,017.54
50	450	727	17.8	8,010	9,307,620	6,766.64
35	315	964	17.2	5,418	6,295,716	6,069.07
22	198	800	16.8	3,326	3,864,812	3,091.85
7	63	638	16.6	1,046	1,215,452	775.46
		4,039			TOTAL/YR	29,480.76

* Two 450-ton machines in parallel

Appendix B: Scheme B: Pump Turbine Thermal Calculations for Type "B"*

Per cent System Load 1	BHP 2	Water Rate lb/bhp hr 3	Steam Rate lb/hr (Col 2 × Col 3) 4	Heat Input Btu/hr (1193 Btu/lb × Col 4) 5	Turbine Heat Drop Btu/lb [2545 Btu/bhp hr ÷ 0.95 × Col 3] 6	Btu/hr (Col 4 × Col 6) 7	Heat in Exhaust Btu/hr (Col 5 − Col 7) 8	Hr 9	Annual Heat (Btu) Drop Input (Col 5 × Col 9) 10	Annual Heat (Btu) Drop (Col 7 × Col 9) 11	Exhaust (Col 10 − Col 11) 12
				80-hp chilled water pump turbine drive							
100	80	47	3,760	4,485,680	57	214,320	4,271,360	32	143,541,760	6,858,240	136,683,520
92	62	47.5	2,945	3,513,385	56	164,920	3,348,465	101	354,851,885	16,656,920	338,194,965
78	39	52.5	2,048	2,443,264	51	104,448	2,338,816	283	689,000,448	29,454,336	659,546,112
64	20	61	1,220	1,455,460	44	53,680	1,401,780	494	718,997,240	26,517,920	692,479,320
50	10	76.5	765	912,645	35	26,775	885,870	727	663,492,915	19,465,425	644,027,490
VI 40	6	90	540	644,220	30	16,200	628,020	2,402	1,547,416,440	38,912,400	1,508,504,040
									4,117,300,688	137,865,241	3,979,435,447
				86-hp condenser water turbine drive							
100	86	47	4,042	4,822,106	57	230,394	4,591,712	32	154,307,392	7,372,608	146,934,784
92	70	49	3,430	4,091,990	55	188,650	3,903,340	101	413,290,990	19,053,650	394,237,340
78	43	55	2,365	2,821,445	49	115,885	2,705,560	283	798,468,935	32,795,455	765,673,480
VI 64	23	64	1,472	1,756,096	42	61,824	1,694,272	3,623	6,362,335,808	223,988,352	6,138,347,456
									7,728,403,125	283,210,065	7,445,193,060

* Reduction gear, variable speed turbines

82 per cent efficiency, the input is 100,000,000 Btu/hr ÷ 0.82 = 122,000,000 Btu/hr. Thus for an output increment of 50,000,000 Btu/hr the incremental input is (122,000,000 – 64,000,000) 57,900,000 Btu/hr. This is equivalent to 50,000,000 Btu/hr ÷ 57,900,000 Btu/hr = 86.4 per cent efficiency. We thus realize a better efficiency for the added increment than for the overall operation.

The foregoing incremental relationship may be expressed as follows:

$$\frac{50}{0.78} + \frac{50}{X} = \frac{100}{0.82}$$

where X = 86.4 per cent

In estimating fuel consumption for steam-actuated equipment in the 900-ton study, an efficiency of 80 per cent is used instead of the 86.4 per cent indicated above. This makes allowance for the small extra power consumed by fans and pumps required in boiler feed-water, fuel and fan-system auxiliaries of boiler operation.

ANNUAL FIXED CHARGE OF BOILER PLANT

Should the annual fixed cost of that portion of an existing boiler plant used for steam air conditioning be charged to the scheme using steam power? The Navy holds that when an existing boiler plant (as in the case under consideration) has ample boiler capacity and requires no expansion – entailing no significant extra operating cost except fuel to supply steam – a charge for interest, depreciation and insurance on the boiler plant is not justified. The steam plant is there, and will be there, regardless of the type of refrigeration equipment installed.

CONCLUSION

The techniques, procedures and approach described herein present a method of evaluating accurately air conditioning refrigeration plant designs for purpose of equipment selection. In the specific application to a Naval 900-ton refrigeration project, the capital worth of annual savings justified the selection of a steam system. The question of maintenance was carefully considered. After weighing all the factors, it was held in this specific project that the difference in maintenance required for the various schemes was negligible.

However, it is conceded that the selection, which was the most economical and logical under the conditions surrounding this installation, may not be the same for other installations having different governing conditions. Much depends upon the factors of availability and type of power; the incremental cost of fuel to make steam, electricity and water, the steam rate of selected turbines; profitable use of by-product steam and heat of condensate; applicable taxes, interest and insurance on capital investment.

The Navy proceeds upon the assumption that every effort should be made to insure the maximum return for the taxpayers' dollars. An economic analysis of air-conditioning operating costs as herein described is required. The procedures demonstrated should prove helpful in guiding contracting engineers to provide evaluations in accordance with these criteria: Service of the required integrity at the lowest total annual cost.

We Are on the Way

S. F. GILMAN
Chairman
Research and Technical Committee

Your Research and Technical Committee has a big job on its hands and this fiscal year, which began on July 1, is proving to be a very busy one. A myriad of details related to the deactivation of the Cleveland Laboratory has had to be studied and decided. The shift to a program of completely sponsored research has had to be made without a Director of Research at Cleveland or a Manager of Research at the New York Headquarters. The needed reorganization of the R & T structure that Bud Palmatier wrote about in the April Journal was tackled by a subcommittee. After several review sessions at R & T Committee meetings, the structure shown on the following page resulted. More about that later.

Deciding upon an organizational structure is one thing; putting it into action swiftly and effectively is another. On July 1, the Technical Committees became responsible for the Society's research program as well as their former technical activities. Consequently, TC membership should include persons with research capability. This was especially urgent for TCs that suddenly had active research projects to guide. As a further complication, to adequately cover the research and technical activities, several changes in TC structure were necessary. Some TCs were split into two or three new TCs. Sometimes, functions of two former TCs were combined into a single TC. Several new TCs were constituted.

The R & T Committee has been deeply concerned that considerable confusion might result and has worked diligently and as swiftly as possible to minimize this confusion. Once the new organization was settled, Chairmen were selected and invited to serve. In some instances, two or more people had to be contacted. Responses to letters varied in speed. Time was of the essence, and the telephone was used as necessary. Here it should be noted that the Headquarters Staff contributed substantially to this effort.

Once a Chairman accepted, how should he proceed? Information and instructions were badly needed. Early in July, a letter went to each TC Chairman showing the new organization grouped into 10 sections, giving brief instructions on how to proceed, plus the name and address of his Section Head. In the meantime, a subcommittee of the R & T Committee was drafting an Operational Guide and last month each TC Chairman received a copy. I say "draft" because the Operational Guide requires approval of the Technical Coordinating Committee and months may elapse before this is properly reviewed and approved. The Guide sets forth the duties and responsibilities of the R & T Committee and of the TCs, and also describes procedures in which TCs will be involved.

The R & T Committee is confident that all TCs are now capable of effective action in handling the research activities.

Although the worst is over, I beg the forbearance of the members during this transition period. About 600 people in former research or technical committees had to be considered for reassignment or reappointment. It is almost a foregone conclusion that some capable people will be lost in the shuffle. Society members formerly on committees who are not notified of a committee assignment in the new organization within the next month are asked to notify the Technical Secretary at the New York Headquarters.

Let's look now at the organization outline on the next page. Although it looks very much like the former Technical Committee structure, there are important differences which should be kept clearly in mind.

As previously stated, the TC responsibility now includes the Society's research program. A TC guiding a cooperative research project will act in much the same manner as a former ASHAE Technical Advisory Committee.

The new 69 Technical Committees are grouped into 10 sections. While the former TCs had Coordinators for the several groups, here each Section is headed by a member of the R & T Committee. Hence, communication with the R & T Committee is direct and action can be fast and effective.

The Section Head has considerable responsibility and latitude in operating his Section. Generally, we expect he will handle all but major problems, although he has the option to refer any matter to the R & T Committee.

The title of the TC is indicative

WHO'S WHO IN ASHRAE

Insofar as possible these listings will each appear twice a year

ASHRAE OFFICERS, DIRECTORS COMMITTEES, STAFF
See page 84, this issue

REGION AND CHAPTER OFFICERS
See revision, next month

RESEARCH AND TECHNICAL COMMITTEES
See page 76, this issue

STANDARDS PROJECTS
See page 84, May JOURNAL

INTER SOCIETY COMMITTEES
See page 84, November JOURNAL

of its scope. In some instances, there may be some overlapping of technical areas or an area may not be covered by any TC. These problems will be solved as experience with the new organization is gained.

Major duties of Technical Committees are:

1. Review and update Guide and Data Book Chapters.
2. Answer technical inquiries submitted to the National Headquarters.
3. Study the research needs in the area covered by the TC.

Organization for Research and Technical Activities

ASHRAE RESEARCH AND TECHNICAL COMMITTEE

S. F. Gilman, *Chairman*
W. T. Pentzer, *Vice Chairman*
H. W. Alyea
F. H. Bridgers
W. L. Holladay
V. J. Johnson
P. J. Marschall
E. P. Palmatier
R. M. Stern
A. G. Wilson
W. F. Wischmeyer
P. Yeomans

TECHNICAL COMMITTEES

SECTION 1.0 FUNDAMENTALS AND GENERAL

V. J. Johnson—Section Head
W. L. Holladay—Asst. Section Head

		Chairman
1.1	Thermodynamics	A. P. Boehmer
1.2	Psychrometrics	J. L. Threlkeld
1.3	Heat Transfer and Fluid Flow	H. A. Whitesel
1.4	Physiology & Human Environment	A. S. Gates, Jr.
1.5	Plant and Animal Physiology	M. K. Fahnestock
1.6	Odor Causes and Control	H. E. Barnebey
1.7	Sound and Vibration	R. E. Parker
1.8	Lubrication	J. D. Bopp
1.9	Refrigerants & Brines	R. C. McHarness
1.10	Corrosion and Water Treatment	B. A. Phillips
1.11	Sorption	W. O. Walker
1.12	Solar Energy Utilization	R. C. Jordan
1.13	Thermoelectricity	E. R. Wolfert
1.14	Owning and Operating Costs	P. N. Vinther
1.15	Instruments and Measurements	F. A. Thomas
1.16	Definitions and Nomenclature	G. P. Priester

SECTION 2.0 HEATING AND COOLING LOADS

A. G. Wilson—Section Head

		Chairman
2.1	Load Calculation Data and Procedures	E. L. Galson
2.2	Weather Data and Design Conditions	W. L. Holladay
2.3	Ventilation Requirements & Infiltration	B. P. Morabito
2.4	Insulation and Moisture Barriers	C. F. Pridmore
2.5	Fenestration	D. J. Vild
2.6	Energy Requirements Estimating	W. S. Harris

SECTION 3.0 HEATING

P. J. Marschall—Section Head

		Chairman
3.1	Hot Water and Steam Heating	A. O. Roche, Jr.
3.2	Warm Air Heating — Fuel Burning	H. G. Hayes
3.3	Electric Heating	G. G. Freyder
3.4	Heat Pumps	G. L. Biehn
3.5	Combustion and Fuels	E. O. Olson
3.6	Industrial Drying Systems	J. R. Thygeson
3.7	Service Water Heating	K. O. Schlentner

SECTION 4.0 VENTILATING AND AIR HANDLING

R. M. Stern—Section Head

		Chairman
4.1	Duct Design and Room Air Distribution	W. W. Kennedy
4.2	Fans	J. B. Graham
4.3	Industrial Environment	J. J. Burke

SECTION 5.0 REFRIGERATION

E. P. Palmatier—Section Head

		Chairman
5.1	Industrial Refrigeration	E. N. Johnson
5.2	Ice Making Plants and Skating Rinks	W. O. Kline
5.3	Ultra-Low Temperature Systems and Test Chambers	D. J. Missimer
5.4	Reciprocating & Rotary Compressor Units	H. P. Soumerai
5.5	Centrifugal Machines	J. R. Chamberlain
5.6	Absorption Machines	L. H. Leonard
5.7	Cryogenics	V. J. Johnson

SECTION 6.0 AIR CONDITIONING EQUIPMENT AND APPLICATION

F. H. Bridgers—Section Head

		Chairman
6.1	Residential & Small Commercial	R. H. Merrick
6.2	Large Buildings	J. R. Duncan
6.3	Industrial	H. E. Ziel
6.4	Transportation	C. W. McElroy
6.5	Room Air Conditioners	P. E. Kolb

SECTION 7.0 FOOD SCIENCE & REFRIGERATION

W. T. Pentzer—Section Head

		Chairman
7.1	Meat, Fish and Poultry Products	K. E. Nielsen
7.2	Dairy Products	E. N. Kerrigan
7.3	Fruits, Vegetables and Other Products	D. H. Dewey
7.4	Frozen Foods	C. F. Evers
7.5	Fermentation Processes & Allied Products	B. H. Bishop
7.6	Concentration, Dehydration & Other Food Preservation Processes	D. K. Tressler
7.7	Cold Storage Warehouses & Locker Plants	W. F. Giacomazzi
7.8	Vehicle and Railway Refrigeration Units	G. A. Gallagher
7.9	Marine Refrigeration	S. W. Brown

SECTION 8.0 DOMESTIC REFRIGERATORS & SMALL REFRIGERATION EQUIPMENT

W. F. Wischmeyer—Section Head

		Chairman
8.1	Domestic Refrigerators & Food Freezers	P. E. Davey
8.2	Beverage Coolers	W. M. Barnes
8.3	Drinking Water Coolers	E. W. Scott
8.4	Ice Makers	D. E. MacLeod
8.5	Commercial Food Storage and Dispensing Equipment	J. H. Rainwater

SECTION 9.0 APPLIED EQUIPMENT

P. Yeomans—Section Head

		Chairman
9.1	Coils, Fan-Coil Units, Air-Cooled Condensers	O. J. Nussbaum
9.2	Liquid Heaf Exchangers	R. M. Armstrong
9.3	Cooling Towers, Evaporative Condensers	W. J. Donovan
9.4	Air Cleaning & Purification Equipment	W. C. L. Hemeon
9.5	Evaporative Coolers	R. S. Ash

SECTION 10.0 AUXILIARY EQUIPMENT

H. W. Alyea—Section Head

		Chairman
10.1	Control	L. F. Flagg
10.2	Refrigerant Flow Controls and Accessories	J. A. Schenk
10.3	Pumps and Piping	W. V. Richards
10.4	Electric Motors—Open and Hermetic	A. P. White
10.5	Safety Devices	F. J. Reed
10.6	Refrigerant Piping	L. N. Brown

TASK GROUPS

	Chairman	Reports to
Task Group on Physiological Research and Human Comfort	R. G. Nevins	TC 1.4
Task Group on Contaminants in Refrigeration Systems	H. O. Spauschus	TC 1.9
Task Group on Hydronics	A. O. Roche	TC 3.1
Task Group on Food Refrigeration and Food Technology	E. J. Robertson	Head Section 7.0
Task Group on Applied Heat Transfer	D. G. Rich	Head Section 9.0
Task Group on Air Conditioning System Control	W. P. Chapman	Head Section 10.0

4. Develop research proposals, determine where the research should be done and how it should be financed.

5. Provide technical guidance to cooperative research projects.

6. Review prospective technical papers.

7. Develop Technical Sessions, Symposiums or Forums for National Meetings.

A Task Group is created for the purpose of accomplishing a special task; when the job is done the Task Group is discharged. It has a great deal of flexibility; for example, it may report directly to the R & T Committee, to the Section Head or to the TC Chairman, depending upon the situation. Typical tasks are to guide an extensive research effort or to explore the research needs in an area broader than a single TC. Six Task Groups have been constituted and you will note that three report to TCs and three to Section Heads.

The Technical Committee will work closely with the Section Head who will assist it in conducting its activities. The R & T Committee envisions each TC eventually supervising at least one research project. Considerable time usually elapses between the initial conception of a research project and the beginning of active work in a research institution. Consequently, it is imperative that each TC study and define its research needs at the earliest possible moment. This is one aspect of the important task ahead if we are to develop a vigorous program of cooperatively sponsored research.

We are not there yet, but we are on the way.

ASHRAE POLICY ON TRAVEL EXPENSE REIMBURSEMENT

Attention of Officers, Directors and of Committeemen is called to the accompanying policy schedule regarding travel vouchers and expenses. Headquarters will supply the appropriate forms in which requests for reimbursements may be made.

The Society recognizes its committee activities as one of real and tangible benefit to the public, the profession and industry interests, and that these are most worthy of proper financial support.

The Society also appreciates the fact that many of the Officers, Directors and committee members do through their own or company resources defray their own travel expenses.

There are, however, situations where expenses due to committee activities would create a hardship on individuals. In such situations the Society is prepared to assist with a contribution toward travel expenses.

Members are always encouraged to bring their wives or families to national meetings, and this practice is particularly encouraged in the case of Officers, Directors and members of the major committees. However, the Society cannot defray travel expenses for this purpose, except as noted in the schedule.

SCHEDULE OF ASHRAE TRAVEL EXPENSE REIMBURSEMENTS

1. *THE PRESIDENT*

a Transportation expenses shall be paid for the President and his wife to all meetings of the Society including Regional meetings and chapter meetings, meetings of other societies for which invitations have been accepted and approved by the Executive Committee in the interest of the Society.

b Mode of transportation shall be at the option of the person or persons travelling and shall be first class air, or its equivalent. Transportation expenses shall include taxi fares to and from point of arrival to hotel or place of meeting and return, and airport bus fares.

c Hotel expenses for the President and his wife including room or suite, meals, and other sundry expenses necessitated by travel.

d Registration fees, and other personal costs in connection with other than Society meetings shall be included as part of the President's travel expense.

2. *EXECUTIVE COMMITTEE*

a The Executive Committee transportation expenses shall be paid to all regularly called meetings and such other meetings as requested by the President or by the Executive Committee. No other expenses will be allowed.

b The mode of travel shall be at the option of the person travelling and shall be that of the minimum air travel rate (i.e., lowest of economy or coach, etc.); or "coach class" rail fare; or automobile mileage of ten cents (10c) per mile—the total not to exceed amount of economy class air travel.

c Registration fees, and other personal costs in connection with other than Society meetings shall be the same as in 1(d) above only when the Executive Committee member is officially representing the President or by authority of the Executive Committee.

3. *REGIONAL DIRECTORS*

a Transportation expenses for Regional Directors shall be paid to all regularly or specially called meetings of the Board of Directors, Regions Central Committee, Chapters Regional meeting in their Region, and other approved regional travel, necessary in the performance of his duty in the interest of the Society, or as directed by the President or the Executive Committee.

b Mode of travel shall be the same as described in 2(b) above.

4. *DIRECTORS AT LARGE*

a Transportation expenses for Directors At Large shall be paid to all regularly and specially called meetings of the Board of Directors, or as directed by the President or Executive Committee.

b Mode of transportation shall be the same as described in 2(b) above.

5. *STAFF TRAVEL*

a All staff travel and expenses shall be subject to the approval of the Executive Secretary in accordance with standard procedures established by the Executive Secretary and in accord with 2(b).

6. *COMMITTEE TRAVEL*

a Members of the following committees who are not included in categories #1 through #4 above, shall be entitled to transportation expenses as defined in 2(b) to all regularly and specially called meetings of their committee:

Advertising	International Relations
Admissions & Advancement	Long Range Planning
Chapters Regional (does not include alternate)	Meetings Arrangements
	Membership Development
Charter & By-Laws	Nominating
Divisional Advisory	Professional Development
Education	Program
Exposition	Public Relations
Finance	Publications
Guide and Data Book	Research & Technical Committees
Honors & Awards	Standards

7. a All expense vouchers must be submitted not ater than thirty (30) days after the meeting for which they are submitted, if payment is requested.

b Any deviations to the above directives are subject to approval of the President or Executive Committee.

8. These directives apply to travel within the Continental United States and Canada (exclusive of Alaska). For those persons serving on committees ex these countries, the maximum allowance would be $300 for any one trip.

Liquefied gases *New Techniques*

Temperature control and the use of liquefied gases

KIRBY M. HAYES

About 13 years ago, research was started on the relation of frozen food quality to temperature. In intervening years, this work has become famous and is known as TTT or time temperature tolerance of frozen foods. The results show that at temperatures above 0 F, quality of frozen food is lost, and that this loss increases at an accelerating rate as temperatures rise. Findings also showed that quality damage accumulates, with the side result being a temperature profile of the industry which pointed up certain deficiencies in the frozen food system. In studies in processing plants, measurements taken in the center cases of a pallet showed that a temperature of 30 F would result in 62 per cent of the high quality life being lost if transferred from the freezing tunnel to 0 F still air before the case temperature reached 0 F.

In a study of frozen food distributors in Massachusetts by a University of Massachusetts team, it was discovered that temperatures of frozen foods being delivered at retail stores were as high as 26 F. No readings were found at zero or below. These temperatures point up the problem of correct storage and display temperatures in retail stores. Neither storage nor display freezers in retail stores are designed to reduce temperatures of frozen foods to 0 F or below quickly, yet in one survey of retail stores, display temperatures ranged from an average low of —4 to an average high of 17 F. These high temperatures can be blamed partly on high delivery temperatures. This indicates that frozen food quality is lowered and that liquefied gases can provide low temperature control.

In this discussion, I am confining my remarks to the two most widely tested gases — liquid nitrogen and liquid carbon dioxide. Each has an extremely high rate of temperature pull-down, has been termed economically usable, and has been tested in various segments of the industry. In order to properly assess their role in the frozen food field, it is necessary to look at the elements within the industry — the processing plant, the warehouse, transportation, distribution and retailing.

Use of liquefied gases within the processing plant can be termed versatile. Estimates by some companies show an extreme reduction in floor space required for the freezing process or unit. These savings in valuable floor space and in construction costs combined with time and money, in the rate of freezing, make the liquid nitrogen freezing system attractive. Processors, however, must meet the minimum production requirements in order to use the system economically.

Such a system, primarily designed as a complete freezing unit, may be adapted to meet two other needs; as a supplement to a regular system at peak periods, and to serve small plants. Adaptations to smaller size can lower prices and at the same time lead to wider use. Areas requiring further investigation are those of plant handling hazards and the effect of extreme low temperatures on packaging materials.

Warehouses are faced with severe temperature reduction problems in the form of "hot loads." Many such loads arrive needing immediate attention to prevent quality loss. Because of prior scheduling, or lack of sufficient space, low temperature blast freezing is not always possible. The alternative is to unload, place in regular storage and spread out as much of the load as possible. A liquefied gas system could easily be installed in a storage room to serve as a temperature reduction room for above 0 F loads.

In the area of frozen food transport, liquefied gases serve a varied and useful purpose. A survey by Quick Frozen Foods showed that, in 1960, approximately 81 per cent of all frozen foods were

Changes are taking place in the overall technology of modern food preservation. New methods have increased in prominence; older techniques are being given new application.

These processes are being accelerated by the continuing growth of the frozen food industry, by the mushrooming of consumer outlets for the

Kirby M. Hayes is Associate Professor in the Department of Food Technology, University of Massachusetts. The paper, excerpted here, was presented as "The Role of Liquefied Gases in the Frozen Food Field" at the ASHRAE 68th Annual Meeting in Denver, Colo., June 26-28, 1961.

es in Food Preservation

moved by truck while 15.5 per cent were moved by rail. In rail transportation, problems are not as involved. Due to more frequent maintenance checks, temperatures in rail shipments of frozen foods are lower than in trucks. More mechanical rail cars are being placed in service with a resultant improvement in temperature control. Where old style rail cars are used or breakdowns occur, liquefied gases can be used to advantage. The knowledge that rapid temperature pull-down is available can improve the entire quality picture. Liquefied gases in initial cooling after loading can boost the speed by which temperature is lowered, and relieve the mechanical unit of this task. Since the newer rail cars have many improvements in insulation, design and adaptability, it is suggested that in times of emergency, a rail car can serve as a temporary storage place utilizing liquefied nitrogen or liquefied carbon dioxide as the refrigerant. One application that must be explored further is the need for portable or truck mounted units located in the yards or at inspection stops to take care of emergency situations.

If liquefied gases become a major part of the frozen food industry, truckers will be the first segment of the industry to accept it.

Problems within the trucking industry are manifold. The job they do is compounded by the fact that both regular truck lines and the independent trucker haul frozen food. Rough roads, minor accidents, careless use of materials handling equipment, all lead to defects in the trailer body. Equipment maintenance is spotty, drivers are not trained to inspect and maintain the thermometers and units.

Other problems involve inefficient loading practices with resultant poor air circulation, poorly insulated bodies, due either to bad design, improper upkeep or punctures. Incorrectly operating units or under capacity units also lead to improper temperature control.

Application of liquefied gases to the frozen food trucking industry has many facets. An application that has been tested and works well is that of liquid nitrogen and liquid carbon dioxide to reduce temperature prior to starting the run. This relieves the refrigerating unit of the heat load and brings the temperature down to levels below 0 F. As the run continues, the unit maintains the proper temperature level. Another application is that of a truck-mounted booster system, utilizing liquid gases and designed to supplement the regular unit. There are, of course, the liquefied gas systems installed in the truck utilizing either liquefied nitrogen or carbon dioxide. These units completely replace the mechanical systems and are operated thermostatically. There are variations of this system whereby, at predetermined stops, further injection of the gases is applied. This eliminates the need for tanks, meters and other equipment on the truck. One useful and almost immediate application of liquefied gases in the trucking industry is locating emergency use units at truck stops to take care of mechanical breakdowns or to supplement truck units.

It is in the field of frozen food distribution that temperature control is most difficult. In a study of distributors made in Massachusetts, the number of frozen food accounts handled by individual firms ranged from 60 to 2000, with most handling between 200 to 500 accounts. The mechanism of servicing these accounts in an orderly fashion, plus the need for filling special orders, increases the problem of temperature control. Where distributors had definite routes, the number varied from 6 to 25 with the average stops per route ranging from 18 to 50. Estimates of the maximum daily mileage per truck was about 100 miles.

Most distributors held the temperatures of their freezers at 0 F. Average product temperatures were 0 F. During loading, product temperatures were observed to vary from no change to a high of 8 F with the rise of product temperature varying according to speed of transfer from warehouse to vehicle. Factors contributing to the temperature rise were: assembly of orders on platform, doors of freezer and truck opened during entire loading period, merchandise allowed to remain on platform for prolonged periods

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end sale of the product and by the overall problems attendant to distribution techniques.

Speakers at the Food Refrigeration Symposium, held at the ASHRAE 68th Annual Meeting in Denver, probed the difficulties inherent in such a growth pattern, concluded that liquefied gases held increasing promise in food preservation usage.

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until transferred to delivery truck. Occasionally, the advantage gained by good handling methods was lost because of failure to pre-cool delivery vehicles adequately or failure to "plug in" refrigeration units while holding overnight.

In checking product temperature at time of delivery, none was found at a temperature of 0 F or below, 22 per cent were between 0 and 5 F, 38 per cent from 6 to 10 F, 27 per cent between 11 and 20 F and 13 per cent above 20 F. Median temperature was 9 F.

CONCLUSIONS

What is the role of liquefied gases in the field of frozen food distribution? Many of the problems outlined earlier are common ones. The first area in which liquefied gases can play an important part is to provide quick cooling for frozen foods received at temperatures above 0 F. For quality maintenance, and to prevent the overloading of the normal storage freezer with warm merchandise, a liquefied gas unit is the least expensive and most rapid way of reducing temperatures to correct levels.

Whether the system used is a complete in-truck thermostatically controlled system, a manually operated booster system, or one used only at dockside to reduce temperature after loading, with the mechanical system taking over thereafter, is not important except to the distributor. The primary purpose of the over-all system is to provide a workable method of maintaining a low temperature in order to protect frozen food quality.

Temperature reduction to levels of –10 to –20 F prior to loading, followed by another gas chilling period to the same level will enable any of the systems described to work well. There is much to be said for the complete in-truck system that can use either a thermostat or a metering device to lower temperature after each delivery. A thermostatically controlled system will be more adaptable to outside temperatures than any other system and have the capacity to meet the cooling need. Liquefied gases can increase turn-around-time and eliminate the need for overnight pre-cooling.

In a final analysis of the role of liquefied gases in the frozen food field, it is evident that liquefied gases will have a definite and useful role in the field from processing plant to retail store door. Economic factors may prevent its use by some companies or segments, yet I feel with further, careful research and development, liquefied gases can provide needed temperature control in the quality maintenance of frozen foods.

Liquid nitrogen as a refrigerant in frozen food transportation

J. J. KANE

Of the many transportation movements required in carrying frozen food to market, the most demanding on a refrigeration system is the final one referred to as multi-stop delivery. Super-markets, restaurants and other receiving points usually maintain a minimum of 0 F inventory space and order in small quantities. This makes it mandatory that the trucker deliver to many such outlets in each run to realize full utilization of truck and driver costs. A 15-ft truck loaded with 600-800 cases may make as many as 40 deliveries in an 8-10-hr day. If the refrigeration system cannot immediately re-establish the 0 F air zone after a delivery is completed it probably never can, since the next door opening allows ambient air to enter again. The net result can be exposing 0 F food to average temperatures only slightly below ambient for protracted periods.

A liquid nitrogen refrigerating system now is available commercially which solves the problems generated because of the factors enumerated earlier. It is based on the use of packaged liquid nitrogen carried aboard the truck.

The key to the operation of the system is the controlled delivery of liquid nitrogen to the truck compartment to maintain temperature at a desired level. The liquid is withdrawn from a vacuum insulated storage container and distributed over the cargo through a spray header. (Fig. 1) An electrically operated valve (solenoid valve) regulates flow to the spray header on an on-off basis as required by a temperature controller. These two units, the liquid solenoid valve and the temperature controller, comprise the only actual moving parts of the system. The system quite literally can be called non-mechanical.

The driving force for delivery of liquid to the spray header is the pressure in the gas phase of the storage container. This pressure comes from the normal evaporation of liquid caused by minor heat leak to the container and residual pressure from the filling operation. During filling, a solenoid valve on the gas phase of the container is open. This provides an escape route for the gas formed from heat leak and cooldown of the transfer lines. With this valve open the pressure in the container is reduced so that the filling of the container can take place more rapidly. When the container is full, the gas solenoid valve is closed. The final pressure in the container will be between 10 and 25 psi. The container pressure is limited to 25 psig by the container relief valve.

The power to operate solenoid valves, pilot lamps and controls is supplied from the 6 or 12-volt truck battery. The electrical schematic wiring diagram, Fig. 2, shows how control of the system is maintained. With the main switch closed, the driver presses reset push-button, energizes relay which is locked into the circuit by its contacts. If the temperature is higher than the desired set point (usually 0 F) the contacts are closed, the liquid solenoid valve is open and liquid flows to the spray header. When a door to the truck compartment is opened, the relay is de-

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J. J. Kane is a Project Engineer in the Cryogenics Products Development Dept., Linde Co., a division of Union Carbide Corp. The paper excerpted above was presented as "Multi-Stop Delivery of Frozen Foods Using Liquid Nitrogen Refrigeration" at the ASHRAE 68th Annual Meeting in Denver, Colo., June 26-28, 1961.

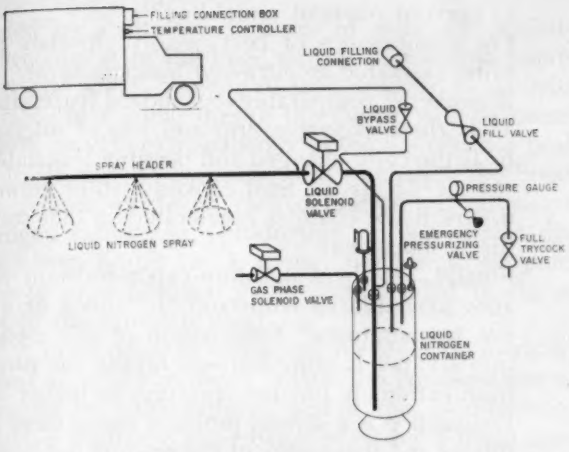

Fig. 1 Multi-stop delivery of frozen food, at desired temperature level, is achieved with liquid nitrogen system shown

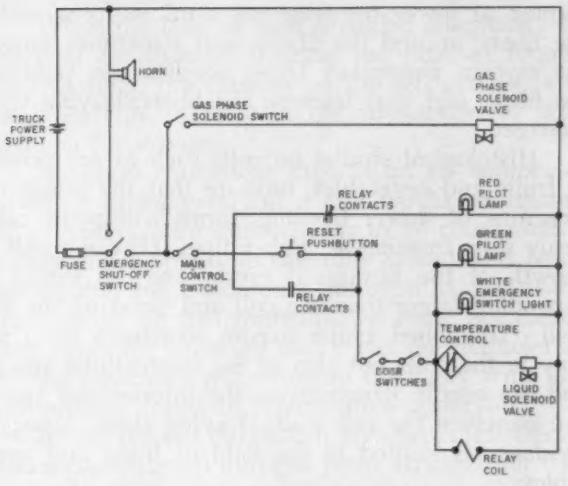

Fig. 2 Schematic outlines details of electrical wiring necessary to control battery powered system

energized by the open door switch preventing the liquid solenoid valve from opening. A red pilot lamp on the truck dashboard becomes lighted as a warning to the driver that the system must be re-energized. An emergency switch located in the truck compartment can be used to prevent the system from operating. Attention is called to this switch being shut off by sounding of the truck horn.

The heart of the system is the liquid container which permits the storage of liquid nitrogen over prolonged periods with almost negligible loss (less than 2 per cent of container per day when the system is not in active operation). The container is vacuum insulated and may have a capacity of 160 or 210 liter depending on the refrigeration requirements. They are designed to ICC specifications using a design working pressure of 150 psig. Each tank has a vacuum safety device as well as the 25-psig liquid container relief valve.

The potential refrigerating capacity of liquid nitrogen is almost 100 per cent realized in food delivery service. Liquid nitrogen converted to a gas and warmed to the temperatures under discussion has an expansion ratio in excess of 600:1. This expansion, coupled with thermal convection currents and the high diffusivity rate of nitrogen into air, results in a quite low temperature gradient throughout the truck. Measurements of the cold air temperature escaping from the truck at the door seals and other leak points have never been further than 10 F from the control temperature. It is generally within 5 F of the interior in the case of trucks in fair condition; the value of this sensible heat loss is negligible. The Btu value of one lb of liquid nitrogen converted to a gas and warmed to 0 F is 165 Btu. The loss of 5 F reduces the utilized portion to 164.0 Btu. Indeed, the difference in refrigerating potential between –20 and 35 F service varies within the narrow limits of 158 and 172 Btu/lb respectively. The controlling factor on liquid nitrogen requirements is the difference in interior and exterior temperatures. Within the experimental accuracy, a straight line function correlates the temperature difference with liquid nitrogen requirements.

Research is needed in ultra-low temperature food freezing.

W. A. MACLINN
Member ASHRAE

With the exception of the paper "The Freezing of Tissues as Seen at the Molecular Level" by B. J. Luyet of the American Foundation for Biological Research, little material is available on the effects of ultra-low temperatures on foods. With the advent of liquid gases for use in food freezing, and the need for engineering applications apparent, the subject is worthy of increased attention on the part of the food scientist.

A molecule of water measures about three angstrom; ice in its simplest form is made up of six molecules of water in a hexagonal figure and these form into a hexagonal prism. This is the ice crystal formed when water or fluid in cells is subjected to temperatures at 0 F or slightly below.

Three other principal crystal structures are formed, depending on the temperature used for freezing. At temperatures in the vicinity of solid CO_2 the structure is irregular dendrites. The basic outside hexagonal form is present but the six water molecules are irregular.

At lower temperatures, but not down to liquid nitrogen, the crystal structure is lobed with radial needles from the center. The hexagonal basic form has become linear in shape. At the lower temperatures of liquid gases, the structure is amorphous and sphere shaped. However, they tend to be transient and form into the radial needle structure.

Histological studies on muscle fibers, such as in meat, fowl, fish and shellfish, indicate that the hexagonal structures, of slower freezing, form on the outside of the fibers, drawing water from within, leaving the fibers intact. The lobed radial needle structures

formed at lower temperatures tend to form within the fibers, around the fibrils, and sometimes around the myosin filaments. These needles can puncture the fibrils and thus fracture the fibers, leaving them damaged.

Histological studies on cells such as are present in fruits and vegetables, indicate that the hexagonal structure, of slower freezing, forms within the cells. Truly slow freezing at higher than 0 F can result in growth of the hexagonal crystals to the extent of expanding larger than the cell and breaking the cell wall. The lobed radial needle structures form not only in the cells but also in the intercellular spaces, and the needle structures in the intercellular spaces can puncture the cell walls, leaving them damaged. Evidence is limited in the field of fruits and vegetables.

The practical experience of this is that the bone in some cuts of meat and in some fish shatters when frozen at ultra-low temperatures. Peas have become extremely friable and crumble when handled.

The segments of the food industry handling the manufactured product needs the following information:

Do ultra-low temperature frozen foods, possibly ruptured by needle-like ice crystals, need especially careful handling in transit to the consumer to prevent physical crumbling?

For maintenance of best quality, do they need to be marketed at ultra-low temperatures?

If marketed comparably to standard frozen foods: a. Is there excessive drip and loss of juiciness?; b. Is the color changed and possibly less stable in storage?; c. Is the final cooked texture harmed?; d. Are flavor changes accelerated so that storage life is shortened?

Finally, are there moisture-vapor resistant packages available to withstand the shock of ultra-low temperatures? Sublimation of cell moisture into a vapor is immediate on raising the product from extremely low temperatures to higher ones. Desiccation is a serious problem under these conditions and the quality of the product can become unacceptable in appearance and texture.

The application of liquid gases to cooling the interior of food transporting vehicles and to reducing the air temperature of loaded vehicles is an actuality in several areas. Unless the injection of the liquid gas is done carelessly, there seems to be no limitation to its use. However, The Refrigeration Research Foundation has found no scientific data to answer the questions posed regarding foods actually frozen by ultra-low temperatures associated with liquid gases.

Merits of gas liquefying cycles in processing and delivery of frozen foods

S. C. COLLINS
Member ASHRAE

Available refrigeration in a liquefied gas is the sum of the latent heat and the gain in sensible heat of the vapor as it warms from boiling point to room temperature. For liquid nitrogen the two parts are about equal in magnitude. Complete utilization of the cooling power of liquid nitrogen in the freezing of foods, therefore, would entail a counter flow process, introduction of a succession of trays of food through a tunnel in one direction while a stream of cold nitrogen flows in the opposite direction. Another method of lesser potential efficiency, thermodynamically, would be to collect the saturated nitrogen vapor and return it to the nitrogen liquefier in such a manner as to recover the refrigerative effect.

Because of the relatively-great power cost of liquid nitrogen as a refrigerant it must be used efficiently. While it is possible to achieve a high degree of effectiveness in the recovery of refrigeration from outgoing cold gas by causing it to flow through incoming material to be frozen, the operation is not a simple one. Heat transfer coefficients between gas and food under likely circumstances are quite low. To compensate for this the tunnel becomes relatively long. Of greater concern is the near impossibility of distributing food uniformly over the cross-section of the tunnel. The result is that the cold gas seeks out channels which permit passage with a minimum of contact with the substance being cooled. Should the food be fluid or semi-fluid in consistency the area of contact with the cold gas stream is limited severely. These factors tend to make the job of recovering refrigeration from nitrogen vapor by direct transfer of heat from food an expensive one.

In summary, the properties of a counter-flow tunnel operation are as follows: 1. Capable, in principle at least, of highest thermodynamic performance or lowest power consumption; 2. Low heat transfer rates and unfavorable distribution of food and coolant streams within the tunnel; 3. Relatively bulky and expensive equipment; 4. Much food in transit and long period of exposure; 5. Not applicable to freezing of food loaded on trucks or cars.

The second method of utilizing nitrogen vapor, namely reliquefaction in a nitrogen plant, is practical and effective although not perfect from a thermodynamic point of view. Warm food would be immersed in or showered with liquid nitrogen with the heat removed from the food being absorbed by the boiling liquid. From the food-freezing compartment the cold nitrogen vapor would be transported in insulated pipes to the heat exchange system of a nitrogen liquefier. Within the heat exchange system the nitrogen vapor would be warmed to room temperature. After compression to 5 atmospheres or more it would be returned to the heat exchanger, cooled and reduced to the liquid state once more.

In essence, method two embraces these factors: 1. Requires slightly more power than method one; 2. Is much more compact. Apparatus is less expensive; 3. In order to be effective, freezer must be accom-

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Samuel C. Collins is Professor in the Department of Mechanical Engineering, Massachusetts Institute of Technology. His paper "Relative Merits of Various Gas Liquefying Cycles for Use in Freezing and Transportation of Foods" was presented at the ASHRAE 68th Annual Meeting in Denver, Colo., June 26-28, 1961.

panied by a nitrogen liquefier of sufficient capacity to accommodate the huge stream of nitrogen vapor. Since the actual freezing of food is likely to be quite variable in volume during the day it will be necessary to provide extra capacity in the liquefier for peak loads. It may be necessary also to change the liquefier regime to favor recondensation of circulated nitrogen rather than production of new liquid distilled from liquid air; 4. It may not always be practical to pipe nitrogen vapor from loaded trucks or cars to the nitrogen plant.

A third possible procedure for freezing foods by means of liquid nitrogen entails the condensation of nitrogen on cold surfaces within the food-freezing chamber itself, the same batch of nitrogen being used over and over again. The refrigerant is gaseous helium circulating in an hermetically sealed system consisting of a compressor, counter-flow heat exchanger, expansion engine or turbine and a condenser. The refrigerating equipment, exclusive of the compressor, would be mounted in such a way that vats loaded with food could be rolled into position immediately below the condenser and held there until the food attains the desired temperature.

Operating routine of a helium refrigerator would be as follows: Beginning with everything at room temperature the compressor and expansion machine are set in motion. During the time required for cooling the heat exchanger and condenser to operating temperature the first load of food is moved into position. Shrouds of canvas or other material are arranged to form a compartment containing the food and the overhead condenser. A small charge of liquid nitrogen is drawn from storage and injected into the compartment for displacing the air. Eventually the condenser becomes sufficiently cold to condense ambient nitrogen and the resultant liquid falls as rain upon the food. In order to keep the concentration of oxygen low, liquid nitrogen from storage will continue to flow in at a low rate but enough to discourage the influx of air.

When the load of food is cooled adequately it is replaced by another. During the interval required for switching, the capacity of the refrigerator is utilized partially or fully in the production of liquid nitrogen for storage.

Auxiliary equipment such as an air compressor, reversing exchangers and a still would be required for providing pure nitrogen gas to be condensed for storage. Such apparatus is readily available. A helium refrigerator for condensing nitrogen is not readily available in the size contemplated here, but there is good cause for believing that it can be produced and that its reliability will compare favorably with more conventional forms of refrigerative apparatus.

In one respect the helium liquefier is less efficient than a conventional nitrogen plant arranged for treatment of a large stream of nitrogen vapor. In both cases the useful refrigerative effect resides in a cold stream of gas from the expansion machine. In the helium refrigerator all of the refrigeration must be absorbed at a single temperature level, namely that of condensing nitrogen. This means that the helium flowing from the expansion machine must enter the condenser several degrees (possibly 40 F) below the boiling point of nitrogen in order to absorb a reasonable amount of heat per lb circulated. Powerwise this is relatively costly. In the conventional nitrogen liquefier the cold expanded nitrogen generally is used to cool a stream of highly compressed nitrogen. The enthalpy of nitrogen at pressures of the order of 30 atmospheres and at temperatures near the critical value changes gradually over a fairly large temperature range rather than precipitously as in condensation at one atmosphere.

In the freezing of food, power consumption by a helium refrigerator would be substantially the same as that of the conventional nitrogen plant. There are possible gains in simplicity of apparatus and ease of operation which must be evaluated carefully in choosing refrigeration equipment for a process of food freezing based on liquid nitrogen.

HTW CENTRAL HEATING PLANT

(*Continued from page 46*)

freeze of the snow melting system. In most cases the control of the amount of heat delivered is achieved by means of throttling water flow through the heat exchangers. At some air heating units in wings and aeroquays of the Air Terminal the control is by diverting a small part of the flow of high temperature water from the coil and back to the Plant. The water returned to the Plant is negligible, and the changes in flow in the entire system are nearly proportional to the heating load.

The lower the return water temperature to the Plant, the less water must be circulated through the system to deliver the same heating load. It is, therefore, one of the aims of the system designer to extract as much useful heat from high temperature water as possible. Controls at heat exchangers were selected so as to approach the temperature of the high temperature water leaving the heat exchanger to the lower temperature of the secondary medium as much as was economical. However, most of the consumers in this system are using low or high pressure steam for the secondary medium, and high temperature water returns cannot be cooled below the steam saturation temperature. In fact, they have to be at least 5 F above that temperature. Since most of the controls at users were provided with means of fine adjustment of the proportional band, rate and reset, the actual return temperatures do not differ from the expected values.

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Program Committee — Plans now nearing final stages tentatively call for four or five technical sessions at the forthcoming St. Louis Semiannual Meeting. It is likely that 16 papers will be scheduled. Thus far, three Symposiums have been decided upon for the 3½-day program.

Board Rule — As presently organized, it is planned that appointments to general committees will be made to totals which are multiples of three; that one-third of the members may be changed each year. By a rule of the Board, the term of membership upon individual general committees is now limited to three years with possible reappointment after one year off.

Presently, there is being assigned to every committee, to safeguard and represent the interests of that committee, a Board member whose duty it shall be to support actively the committee's recommendations to the Board.

The *Long Range Planning Committee*. Upon recommendation of this committee, it has been made a rule of the Board to permit chartering additional chapters within a geographical area, providing that the membership assigned to an existing chapter is in excess of 500 and that a petition for a new chapter be signed by not less than 100 members in the chapter area.

This is based upon the recognition that, owing to falling attendance ratios and other problems in chapters in large metropolitan areas, establishment of more than one chapter can be justified in such areas to enable fulfillment of the objectives of the Society.

General Administrative and Coordinating Committee has recommended consideration of a plan to issue an annual report of the Society to be distributed to both members and selected non-members. Included would be a statement by the President and research and financial statements.

Research and Technical Committee — Of the 66 Research Committees, chairmen for 57 had been appointed at the time of the Denver Meeting.

Approval of the R&T Committee of a grant of funds to the University of Florida for a study of residential heating and cooling loads has been accorded, dependent upon the completion of the residence and the availability of adequate instrumentation there. The committee has recommended to the Board also that $6000 be granted the University of Illinois for research in steam and condensate flow.

All of the senior staff at the Laboratory has relocated. W. F. Kerka has joined the staff at Fenn College. L. F. Schutrum is now with the Pittsburgh Plate Glass Company. R. G. Huebscher is associated with Thompson Ramo Woolbridge, Inc. Ilse Jahn will head the German Department at Shaw High School.

Board of Directors — 1961 ASHRAE TRANSACTIONS will be published in book form, as in previous years, and be made available to members for $3; to non-members for $6. Edition to be limited, as it was for 1960.

At Headquarters — By the time this issue of the JOURNAL is published, the headquarters operations of the Society in New York will have been consolidated at our new Headquarters in the United Engineering Center, 345 East 47th Street.

As reported elsewhere in this issue, former Technical Secretary A. T. Boggs III is now Associate Secretary. Former Assistant Director of Research C. M. Humphreys is the new Technical Secretary of ASHRAE.

Meetings ahead

September 20-22—International Institute of Refrigeration, Commissions 2, 3, 6A, 6B and 8, Cambridge, England.

September 27-October 4 – Institution of Heating and Ventilating Engineers, International Heating, Ventilating and Air Conditioning Conference, London, England.

October 2-4 – American Gas Association, Annual Convention, Dallas, Texas.

October 7-10 – Western Building Industries Exposition, Los Angeles, Calif.

October 23-27—National Metal Exposition, Detroit, Mich.

October 31-November 2 – Fourth Canadian Refrigeration and Air Conditioning Show, Toronto, Ont.

November 5-7—National Frozen Food Association, National Convention and Exposition, Bal Harbour, Fla.

November 6-10 – National Warm Air Heating and Air Conditioning Association, 48th Annual Convention, Chicago, Ill.

November 12-15 – Air Conditioning and Refrigeration Institute, Annual Meeting, Hot Springs, Va.

November 13-17 – National Electrical Manufacturers Association, Annual Meeting, Atlantic City, N. J.

November 26-December 1—American Society of Mechanical Engineers, Winter Annual Meeting, New York, N. Y.

November 28-30 – Building Research Institute, Fall Conferences, Washington, D. C.

January 29-February 1—American Society of Heating, Refrigerating and Air Conditioning Engineers, Semiannual Meeting, St. Louis, Mo.

February 9-11—Air Conditioning and Refrigeration Wholesalers, Annual Convention, Los Angeles, Calif.

February 12-15—12th Exposition of the Air Conditioning, Heating and Refrigeration Industry, Los Angeles, Calif.

News of ASHRAE members

Honors and Recognitions

Anthony Hass, President of Dryomatic Corporation, has been elected President of the Greater Washington Industrial Council. Vice President of the Council for the past year, having previously been its Secretary, he has been connected with Dryomatic since 1950, first as Vice President and Sales Manager. In addition to membership in ASHRAE, he is Vice Chairman of the Adsorption Dehumidifier Section of Nema, and a member of the American Chemical Society and American Ordnance Association.

New jobs

Charles C. Hansen III, Chief Engineer of Refrigerating Specialties Company for the past eight years, has been elected President. He had previously served in various refrigeration engineering positions with Burge Ice Machine Company, Betz Corporation and Liquid Carbonic Corporation. A graduate of Cornell University (BME 1946), he also attended Massachusetts Institute of Technology (MS 1947) and the University of Chicago (MA 1955).

Frank E. Parsons has been appointed by Allen Industries, Inc., to direct national sales of their polyurethane foam products. Bringing to the company experience with all types of thermal insulations, he has pioneered and developed new products and, as Industry Engineer for the National Mineral Wool Association, he had been active in improving product and application standards.

Donald G. Harter is now Vice President—Research and Engineering, for Kodiak, Inc. Prior to joining the company he was Director of Engineering for York Div of Borg-Warner Corporation, having joined that organization in 1958. He came to York after ten years with the Kelvinator Div of American Motors Corporation, where he served as Chief Engineer. In addition to ASHRAE, he is a member of the Society of Automotive Engineers, American Ordnance Association and American Society For Metals.

G. Dale Mericle, for 17 years Associate Editor of Air Conditioning, Heating & Refrigeration News, has been named General Manager of the recently reorganized Technical Book Div of Business News Publishing Company. Editor of many of the service manuals published by the company and author of a four-volume series on automobile air conditioners, he is a graduate of Wayne State University. He will continue his association with the News.

Newton W. Mac Nichol has joined Bell & Gossett Company as a special field representative for heating and cooling products in the company's Eastern Div. He was formerly General Sales Manager for Warren Webster & Company.

Peter J. Dalton has been appointed Manager of Sales Training for Dunham-Bush, Inc., and will assume responsibility for the company's Education Center program, field training and engineering college liaison. Most recently Vice President, Sales, at Welbilt Air Conditioning and Heating Corporation, he was previously Product Sales Manager for Worthington Corporation. A graduate of the University of Connecticut, he received a master's degree in mechanical engineering from the University of Miami.

William P. Myers recently joined the Trane Company, Memphis, Tenn., sales office. He will specialize in the sale of packaged air conditioning and heating equipment.

T. M. Broderick has been named Manager of motor sales for the Hunter Div, Robbins & Myers, Inc., leaving his former post with the Delco Products Div of General Motors Corporation.

Mark Mooney has joined the staff of WeatherKing of Florida, Inc., as a manufacturers' representative. Vice President of Florida West Coast Chapter of ASHRAE, he was for many years Product Manager at Carrier Corporation, later serving as Vice President of Sales, Typhoon Air Conditioning Div, Hupp Corporation.

W. A. Edwards, newly appointed Sales Representative for the OEM Coil and Commercial Refrigeration Departments of the Danville Div of Bohn Aluminum & Brass Corporation, will represent that company in the state of Georgia and eastern Florida.

Thomas E. Scott, a sales engineer in the Albany office of Trane Company, becomes Manager of the newly opened sales office in Poughkeepsie. He is a mechanical engineering graduate of Purdue University.

Robert K. Moss, former Vice President of Dell Corporation, has announced the formation of his own consulting organization, Robert K. Moss, Consultants. He has had more than fifteen years of experience in the fields of heating and air conditioning.

Dean C. Seitz, Sales Manager of the Wholesale Refrigeration Department of McQuay, Inc., has been appointed to the additional post of Vice President – Marketing of American Automatic Ice Machine Company, a subsidiary of McQuay.

Joseph H. Maloney, appointed sales agent for General Blower Company in the Louisiana, lower Mississippi and Florida Gulf Coast areas, brings to the company more than eleven years' experience in fan and blower application engineering.

Jack Morgan will handle both heating and cooling products in the Maryland area for Dunham-Bush, Inc. He was formerly Project Manager with Lloyd E. Mitchell, Inc.

Myron E. Hunzeker has been promoted to Manager of the Trane Company sales office in Peoria, Ill., advancing from his post as Sales Engineer. He received degrees in mechanical and electrical engineering from Iowa State University.

Necrology

Joseph T. Gier, Associate Professor of Engineering, University of California, died recently at the age of 51. He had been a member of the Society since 1954.

Dale G. McCauley, retired President of Lansing Heating and Ventilating Company, died this year at the age of 59.

Charles F. Newport, an 87-year old Life Member, died June 26 in a convalescent home where he had lived for the past year and a half. He retired 15 years ago after 25 years with Weil-McLain Company as a heating and sales engineer.

Stanley E. Plewes, a Life Member who was elected to membership in 1917, died recently at the age of 76.

Stanley Dreskin, a heating and ventilating engineer with Syska & Hennessy, died in June. He first joined the Society as a Junior Member in 1949, advancing to Associate in 1956.

Adolph Gustav Syska, Consulting Engineer and a former Partner in the firm of Syska & Hennessy, died recently at the age of 72. He had been a Member since 1933.

BULLETINS

Central Air Conditioners. In addition to general and mechanical design features, 60-page Catalog 560 offers detailed, step-by-step information to assist in selection of unit type and size, cooling and heating coils, accessories and determination of fan speed and motor hp requirements for this line of central air conditioners. Dimensional data also are tabulated.
McQuay, Inc., 1601 Broadway St. N.E., Minneapolis 13, Minn.

Insulation. Various types of insulation, for use at high, intermediate, ice water and frigid temperatures and for heating, air conditioning and low pressure steam are described in four-page Bulletin SC-9/0. Among materials covered are mineral wool pipe covering, industrial felt, expanded polystyrene, cork, glass fiber, calcium silicate and combinations of calcined diatomaceous earth and asbestos.
Mundet Cork Corporation, Insulation Div, 7101 Tonnelle Ave., North Bergen, N. J.

High Vacuum Vapor Pumps. Partial contents of 56-page Bulletin 6-1 are: performance characteristics of vapor pumps, metal oil pumps, metal mercury pumps, glass oil pumps, glass mercury pumps, pump and port assembly, adapters and pump fluids. Tables, charts and photographs accompany the text.
Consolidated Vacuum Corporation, a Subsidiary of Consolidated Electrodynamics/Bell & Howell, Rochester 3, N. Y.

Prefabricated Intake-Exhaust Unit. For handling ventilation problems through the roof, the Extruded Aluminum Penthouse is weatherproof in design, with louver blades extending continuously around all four sides. Where air control is desired, operating louvers can be supplied in place of the standard fixed blades. Descriptive and specification data are presented in 20-page Catalog C-S L-1160.
Construction Specialties, Inc., 55 Winans Ave., Cranford, N. J.

Fans. Construction features of HS Fans, Series 81, Class I and II, are discussed in 44-page Bulletin A-1203, which also describes available drive arrangements, configurations and types of drive. Specific components, such as bearings, dampers, vanes, sound-absorbing bases and drive speed control devices are covered also.

(*Continued on page 96*)

Candidates for ASHRAE Membership

Following is a list of 73 candidates for membership or advancement in membership grade. Members are requested to assume their full share of responsibility in the acceptance of these candidates for membership by advising the Executive Secretary on or before September 30, 1961 of any whose eligibility for membership is questioned. Unless such objection is made these candidates will be voted by the Board of Directors.

Note: * Advancement † Reinstatement

REGION I

Connecticut

CAVANAUGH, C. T.†, Dist. Sales Mgr., Dunham-Bush, Inc., West Hartford.

Massachusetts

MCCLEAN, F. C.*, Treas. & Owner, Fred C. McClean Heating Supplies, Inc., Springfield.
TARLOW, L. E., Project Engr., Metcalf & Eddy, Boston.

New Jersey

CAILLE, E. H.*, A-C. Engr., Montgomery Engineering Co., Jersey City.
HEINTZ, C. J., Engr., Superior Air Conditioning, West Orange.
ZIMMER, P. C., Project Engr., Kramer Trenton Co., Trenton.

New York

ELLARD, P. R.†, Service Engr., Carrier Corp., Syracuse.
GREELEY, E. M., Sr. Chem. Engr., Carrier Corp., Syracuse.
KORMAN, J. F., Repr., Johns-Manville Sales, New York.
MORCATE, J. L., Plant Engr., Breakstone Foods Div., New York.

REGION II

Canada

BRADFIELD, K. R.*, Apprentice Co-Ordinator, The McKinnon Industries Ltd., St. Catharines, Ontario.
DAYUS, L. G., Pres. & Gen. Mgr., Lloyd Register & Grille Co., Ltd., Islington, Ontario.
DEBLOIS, T. M., Sales Engr., Canadian General Electric Co., Ltd., Toronto, Ontario.
FALKE, J. F.†, Chief Engr., Copeland Refrigeration of Canada, Ltd., Brantford, Ontario.
FORTIER, P. G.*, Gen. Mgr., Canadian Comstock Co., Ltd., Ottawa, Ontario.
HOOKER, R. G., Sales Mgr., Vibron Limited, Toronto, Ontario.
KOROL, PETER, Repr., Hagan Corp. (Canada) Ltd., Toronto, Ontario.
MATTHEWS, J. R., Repr., Martin Brothers, Ltd., Rexdale, Ontario.
PATTON, W. J., Chief Engr., Silverline Mfg. Co., Ltd., Winnipeg, Manitoba.
WYLIE, D. J., Sales Mgr., Marley Canadian Ltd., Toronto, Ontario.

REGION III

District of Columbia

SHEA, J. E.*, Chief Mech. Engr., Thomas B. Bourne Assoc., Inc., Washington.

Maryland

CHASON, E. R., Service Repr., Chason Service Eng., Inc., Baltimore.

Pennsylvania

CORBMAN, A. L.†, Vice-Pres. & Secy., Corbman Bros., Philadelphia.
JOHNSON, R. E., Self-Employed, Springfield.
LEDGER, J. A., Lab. Tech., John J. Nesbitt, Inc., Philadelphia.

Virginia

CALVERT, R. W., Service Mgr., Wash. Refrigeration Service Co., Arlington.
MORSE, R. F., Partner, Brandt & Morse, Richmond.
TARRANT, R. L., Sales Repr., The Trane Co., Norfolk.
WATSON, H. L.*, Htg. & A-C Engr., Refrigeration Supply Co., Richmond.

BULLETINS

Industrial Valves. Added to this line are bronze, solder-end gate valves, covered in Bulletin 1013. These 150-psi WSP, 300-psi WOG valves, designed for use with types K, L and M copper tubing, are available in both rising and non-rising stem design in sizes from ¼ through 2 in.
Ohio Injector Company, Wadsworth, Ohio.

Acoustic Panels. Designed for use as sound barriers and partial enclosures for control of noise in plants and offices, Noishield panels are the subject of Bulletin C70. Each panel is of rugged construction and is available in cold rolled steel, galvanized steel, stainless steel and other materials for specific needs.
Industrial Acoustics Company, Inc., 341 Jackson Ave., New York 54, N. Y.

Exhaust Systems. How to handle corrosive fumes with fans, air washers and exhaust systems made of polyvinyl chloride (PVC) is described in 36-page Catalog F. Photographs, dimensional diagrams, performance tables and description of each item are given separately. Design and applications are presented and chemical-resistance data are tabulated for normal and high-impact PVC and various metals.
Industrial Plastics Fabricators, Inc., Endicott St., Norwood, Mass.

Foam Insulation. Descriptive of Hewfoam, a new rigid urethane foam insulation, is an illustrated flyer. Cited for the product are solvent, heat and moisture resistance and excellent insulating qualities. Pointed out are characteristics of the material, such as its light weight and sound absorption ability.
H. E. Werner, Inc., 2990 Industrial Blvd., Bethel Park, Pa.

REGION IV

Florida

DORINI, D. K., Field Sales Engr., Acme Industries, Inc., Fort Lauderdale.

REGION V

Indiana

STRUBEL, C. W., Associate, Wm. H. Kashner & Assoc., Lafayette.

Ohio

BREZINA, W. G., Mech. Designer, C. E. I. Company, Cleveland.
HUMPHREVILLE, T. N., Engr. Section Mgr., Westinghouse Electric Corp., Cleveland.
HUNGERFORD, E. N., Engr., Cleveland Electric Illum. Co., Cleveland.
KOLANZ, R. P., Sales Engr., General Electric Company, Cleveland.
RADLE, W. J.*, Performance Analyst, Airtemp Div., Chrysler Corp., Dayton.
SMITH, C. H. JR., Project Engr., The Kruger Co., Cincinnati.

REGION VI

Illinois

LEHNERS, R. W., Cons. Engr., Joliet.
SHANTZ, T. A., Sales Engr., Marc A. Shantz, Chicago.
STREB, G. W.*, Secy., Sherman Plumbing & Heating Inc., Crystal Lake.

Michigan

ADAMS, R. E., Engr., Oldsmobile Div., General Motors Corp., Lansing.
BLOY, F. G., Sales Mgr., Wall Tube & Metal Products, Detroit.
SMITH, J. E.*, Field Service Repr., Whirlpool Corp., Benton Harbor.

REGION VII

Tennessee

SELLICK, J. P., Bus. Dev. Engr. Program Mgr., Nashville Div., Avco Corp., Nashville.

REGION VIII

Oklahoma

RICHARDSON, W. H.†, Vice-Pres., Engineering for Industry, Oklahoma City.

Texas

LAKE, S. T. JR.*, Br. Mgr., J. R. Dowdell & Co., Ft. Worth.
ROYAL, L. R., Sales Repr., The Trane Co., Dallas.
WARD, W. H. Jr., Engr., Texas State Bldg. Comm., Austin.

REGION IX

Colorado

STEWART, R. B., Supvsg. Mech. Engr., Cryogenic Engineering Lab., Boulder.

Nebraska

UPHAM, R. WM.†, Chief Mech. Br. A-C. & Refr. Engr., HQ Strategic Air Command, Offutt AFB.

South Dakota

BERGMAN, BARNEY, Repr., Amstan Supply, Sioux Falls.
GRANTIER, A. J., Sales Engr., Johnson Service Co., Sioux Falls.

REGION X

Arizona

CULLINANE, J. J., Owner, Cullinane Sales Co., Phoenix.
KIMBALL, C. T., Sales Engr., National Refrigeration Sales, Phoenix.

California

CHUN, V. K., Plant Staff Engr., Hughes Aircraft Co., Culver City.
CUTHBERT, B. F., Mech. Designer, Calif. Institute of Technology, Pasadena.
WERNER, J. E., Sales Engr., The Trane Co., San Francisco.
WIGHTMAN, G. W.*, Mech. Engr., Ayres & Hayakawa, Los Angeles.

Washington

TANEMURA, K. T.*, Research Engr. B, The Boeing Co., Seattle.

FOREIGN

Argentina

BOYD, J. C., Sr. Partner, De Weerth Boyd & Assocs., Buenos Aires.

Holland

WENT, ADRIAAN, Tech. Man. Director, Marynen Koelindustrie N.V., The Hague.

India

ARORA, C. P., Lecturer, Indian Institute of Technology, Powai, Bombay.
MALLAPUR, S. V., Reader, Principal Engg. College, Gulbarga.
RAO, H. V., Teacher Trainee, Indian Institute of Technology, Khargpur, W. Bengal.

Malaya

LEE, M. B. P., Lecturer, University of Malaya, Pantai Valley, Kuala Lumpur.

Mexico

MARTINEZ, E. A.†, Project Engr., Manufacturers General Electric, S.A. De C.V. Mexico.
SCHEIDER, OTTO, Exec. Gen. Mgr., Grasso De Mexico, S.A., Mexico.

New Zealand

STEAD, R. S., Mgr., Wilkins & Davies Construction Co., Ltd., Lower Hutt.

Pakistan

ALI, S. Z., Resident Engr., State Bank of Pakistan, Karachi.

Scotland

HOLDEN, STANLEY, Asst. to Joint Managing Dir., Thermotank Ltd., Glasgow.
WATSON, H. J., Gen. Mgr., Engg. Div., Thermotank Ltd., Glasgow.
WU, A. B. L., Trainee Engr., L. Sterne & Co., Ltd., Glasgow.

STUDENT

MALIK, D. P., Milwaukee School of Engineering, Milwaukee, Wisconsin.

IHVE Meeting in London This Month

Among those ASHRAE members planning to attend the International Heating, Ventilating and Air Conditioning Conference in London, England, September 27-October 4, are President and Mrs. John Everetts, Jr.

Conducted under the auspices of the Institution of Heating and Ventilating Engineers, whose President Leslie Rayner attended our Semiannual Meeting in Chicago earlier this year, the Conference has been projected toward those administrative and technical advances likely in the next 10 years, as well as to the integrated design of architectural and engineering services for economy of building construction.

Concurrently with the Conference, there will be the first International Heating, Ventilating and Air Conditioning Exposition at Olympia in London.

Jointly cooperating with IHVE in this undertaking are the Association of Heating, Ventilating and Domestic Engineering Employers and the Heating and Ventilating Research Association.

Penthouse of the library holds boilers, chiller, fans and pumps for entire building, leaving the basement free for book storage

30-Zone Conditioning System Cools New Library

Minneapolis now has an $8 million library-planetarium of distinctive air conditioned characteristics to replace a 70-year outmoded structure. The four-story library section contains space for some two million books and 800 patrons, plus reading areas, meeting rooms, library offices, employee lounges and a cafeteria. A single-story structure next to the library houses a 200-seat planetarium, a 450-seat auditorium, a natural-science museum, displays and workshop areas.

Although the two buildings are connected by an entrance arcade, they are served by separate air conditioning systems having a single chiller. For the library, all mechanical equipment is located in a fifth floor penthouse. By putting the fans, boilers and compressor on top of the building, the two basement levels are made available for storing books.

Notable among system features is the provision of eight air handling layouts for the planetarium wing and the provision of no less than 30 zones in the auditorium, planetarium, galleries, museums and other areas.

Conventional induction units wipe the library windows with temperature-controlled air (at about 60 F), with wall-mounted thermostats controlling water flow through the units, adding more heat in winter, more cooling in summer. Air movement is from these perimeter units to large return air grilles in the center of the building. These grilles open onto a common vertical airshaft going to a 150,000-cfm return air fan in the penthouse.

Return air either is exhausted or recirculated through the building, depending on outdoor temperature. Automatic dampers keep the building interior at a ¾-in. static pressure. The supply system uses four fans (two 65,000-cfm, two 13,000-cfm) to shunt air to the perimeter units. Incoming air first passes through preheat coils, then mixes with return

Plenum mixes outside air (from dampers at left) with recirculated interior air, then feeds it to individual air-handling systems through dampers at right

Books from old building (which had heating only) absorbed enough moisture to reduce humidity readings 45 to 30% in one day

air from the building interior. Next, mechanical filters and electronic air cleaners clean the air before it passes over chiller coils. For humidity control, the air goes through cooler coils (a 300-gpm pump supplies well water for these coils) and then over reheat (hot-water) coils.

Chiller coils are fed by a 500-ton unit powered by a 490-hp motor. The unit is mounted on a spring-loaded platform, as is all mechanical equipment in the penthouse. The cooling tower is screened by a vertical anodized aluminum grille; from the outside the entire penthouse wall is unbroken. For additional efficiency the outside air intake is behind the cooling tower, assuring air flow even on still days. The cooling tower is also fitted with a 20-hp fan.

Heat for the building is provided by two 150-hp gas-fired boilers, with oil standby. Domestic hot water is furnished by a 500-gal water tank, the water heated to 145 F. A two-ton chiller cools the drinking water for the institution.

While the air conditioning system can control heat and humidity in the library, it can do little about the dust and dirt of Minneapolis. This problem was solved with the installation of Minneapolis-Honeywell electronic air cleaners in each air handling system in both the library and the planetarium. Because the electronic air cleaners remove over 90% of the dirt, the conditioned air can be recirculated over and over again, thereby greatly reducing the percentage of outside air.

HEAT AND MASS TRANSFER ON COILS

(*Continued from page 54*)

quite small and the accuracy of the measurement was not sufficient for conclusions.

This research project is continuing and similar data for an extended surface coil with the addition of friction data should be available in the near future.

ACKNOWLEDGMENT

The author gratefully acknowledges the financial assistance from the ASHRAE. Acknowledgment also is made to Karl Ko-Yi Chen, former graduate student at Case Institute of Technology, for obtaining most of the experimental data presented in this paper. Work on this research project is continuing under a research grant from ASHRAE.

NOTATIONS

DB Dry bulb temperature of the air, F
WB Wet bulb temperature of the air, F
t_s Surface temperature of the coil, F
Q_s Rate of sensible heat transferred, Btu/hr
Q_L Rate of latent heat transferred, Btu/hr
M_T Rate of mass of water transferred, lbm/hr
A Heat transfer area of coil, ft²
h Coefficient of heat transfer from air to surface, Btu/hr ft² F
M_a Mass rate of flow of "dry" air, lbm/hr
C_h Humid specific heat, Btu/lbm R
ln Napierian or natural logarithm
h_D Coefficient of mass transfer, lbm/hr ft²
W Humidity ratio, lbm water/lbm air
V Coil face velocity of the air, fpm
L.R. Lewis Relation, $\frac{h}{h_D c_h}$
λ Latent heat of vaporization, Btu/lbm

APPENDIX

Over a differential element of a coil surface, the air changes temperature by ΔDB and at the same time is reduced in moisture content by ΔW. The total heat transferred to a surface element, dA, is the sensible heat from the air and from the water vapor in the air and the latent heat equivalent of the water condensed from the air

$$Q_T = Q_s + Q_L \qquad (a)$$

The sensible heat transferred is

$$Q_s = hdA\,(DB - t_s) = -M_a\,c_h\,dDB \qquad (b)$$

The humid specific heat, c_h, is used to account for both the cooling of the air as well as for the cooling of the water vapor in the air. The desuperheating of the water condensed on the differential element is

$$M_a\,c_v\,(DB - t_s)\,dW$$

and the equation for the sensible heat transferred at the differential element is:

$$h\,(DB - t_s)\,dA = -M_a\,c_h\,dDB - M_a\,c_v\,(DB - t_s)\,dW \qquad (c)$$

The latent heat or mass of water condensed on the differential element is

$$\lambda Q = h_D\,(W - W_s)\,dA = -M_a\,dW \qquad (d)$$

By combining equations c and d the ratio of the mass change to temperature change can be obtained

$$\frac{dW}{dDB} = \frac{c_h}{(DB - t_s)\,\frac{h}{h_D} \cdot \frac{1}{W - W_s} - c_v} \qquad (e)$$

For a constant surface temperature this equation can be integrated over the total coil surface. Note that the specific heat of the superheated steam in the air c_v can be represented as

$$c_v = \frac{c_h - c_s}{W} \qquad (f)$$

Making this substitution and integrating from initial state 1 to final state 2 gives:

$$\frac{h}{h_D c_h} I_n \left(\frac{W_2 - W_s}{W_1 - W_s}\right) - \left(\frac{c_h - c_s}{c_h} \cdot I_n \frac{W_2}{W_1}\right) = I_n \left(\frac{DB_2 - t_s}{DB_1 - t_s}\right) \qquad (g)$$

It can be seen that the second term on the left contributed by the desuperheating of the condensed water is negligible. Making this simplification, the equation reduces to:

$$\frac{DB_2 - t_s}{DB_1 - t_s} = \left(\frac{W_2 - W_s}{W_1 - W_s}\right)^{\frac{h}{h_D c_h}} \qquad (h)$$

It is to be noted here that the ratio of the temperature differences is equal to the ratio of the humidity ratio differences to an exponential power. The exponent is the Lewis Relation. The locus of the air temperature at any point, x, in the coil is then given by Equation 9 in the paper.

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STAFF CHANGES

To improve Headquarters effectiveness and services and to adjust to the closing of the ASHRAE Research Laboratory, two major staff changes are announced currently.

A. T. BOGGS III

Andrew T. Boggs III, formerly Technical Secretary, will become Associate Secretary to assist Executive Secretary R. C. Cross in his various duties. Presently, Mr. Boggs will concentrate upon Membership and Chapter relationships, including the currently inaugurated add-2000-members campaign, and will undertake certain special studies of operational factors. He will continue to serve as liaison for the Standards and Program Committees until his successor may do so.

Mr. Boggs, who has been ASHRAE Technical Secretary since the ASHAE-ASRE merger, also had served with the latter organization in that capacity since October 1955. His activities have included staff administration of the work of the Program, Standards, Education, and Research Exhibit Committees, Technical Committees of the Research and Technical Committee and Technical Coordinating Committee.

He has been ASHRAE representative on the ASA Conference Advisory Board, Standards Council Nominating Committee and Alternate on the Standards Council. He is Secretary to the U. S. Advisory Committee for ASA on international standards concerning refrigeration.

Mr. Boggs is Secretary of ASA Sectional Committees: B9 Safety-Code for Mechanical Refrigeration, Interpretations Subcommittee; B38 Household Refrigerators and Freezers; B59 Practice for Mechanical Refrigeration Installation on Shipboard; B60 Refrigerant Expansion Valves.

Prior to joining this Society, he was assistant to the Assistant Managing Director of the Edison Electric Institute, undertook special studies and was secretary to the Joint Heat Pump Committee.

Mr. Boggs, a veteran of World War II, is a Major in the Reserve Program and served with the 8th Armed Division in Europe. He was awarded the Belgium Military Cross. A resident of Westfield, N. J., Mr. Boggs is Vice President of the Garden State Chapter National Sojourners and a Deacon of the First Baptist Church.

Clark M. Humphreys who has, for 7 years, served as Assistant Director of Research at the ASHRAE Laboratory in Cleveland will, upon the completion of terminative operations there, report to New York Headquarters as Technical Secretary. Mr. Humphreys will actively coordinate staff work upon National Meetings Programs with the Program Committee, will provide liaison with the Standards Committee and take over other functions of the Technical Secretary.

ASHRAE OFFICERS, DIRECTORS COMMITTEES, STAFF
See page 84, this issue

REGION AND CHAPTER OFFICERS,
New listing, next month

RESEARCH AND TECHNICAL COMMITTEES
See page 76, this issue

STANDARDS PROJECTS
See page 74, January JOURNAL

C. M. HUMPHREYS

Mr. Humphreys was graduated from Ohio State University in 1925 with a Bachelor's degree in Mechanical Engineering. In 1942 he was awarded a professional M.E. degree from the same school.

From 1925-1931, Mr. Humphreys was employed as Mechanical Engineer in the office of the State Architect and Engineer in Columbus, Ohio, where he had charge of the design of heating and plumbing systems for buildings being built at the various state institutions. From 1931-1944 he was Assistant Professor of Mechanical Engineering at Carnegie Institute of Technology, and Head of the Plumbing, Heating, and Air Conditioning Option. During this period he was also active in consulting work, primarily in the heating design field. In 1944 he accepted a position as Mechanical Engineer in the Industrial Architectural firm of Hunting Davis & Dunnells in Pittsburgh, Pa.

In 1945 Mr. Humphreys came to the ASHRAE Research Laboratory as Senior Engineer under Director Cyril Tasker. In 1954 he became Assistant Director at the Society Laboratory and has continued in this capacity up to the present.

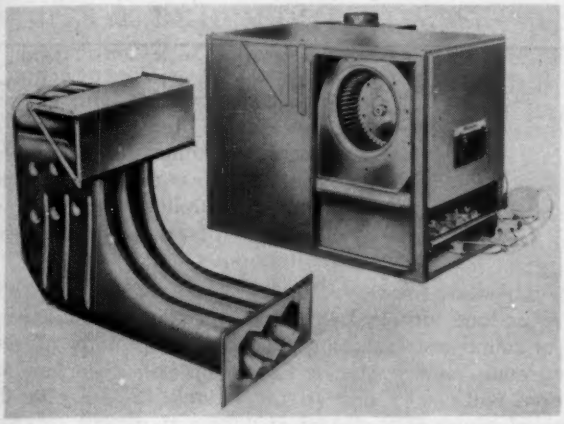
ASHRAE NATIONAL MEETINGS AHEAD

1962	1963	1964
Jan. 29-Feb. 1 Semiannual St. Louis, Mo.	Feb. 11-14 Semiannual New York, N. Y.	January 27-29 Semiannual New Orleans, La.
June 25-27 69th Annual Miami Beach, Fla.	June 24-26 70th Annual Milwaukee, Wisc.	June 29-July 1 71st Annual Cleveland, Ohio

PARTS AND PRODUCTS

COMPACT FURNACE

Suited for installation in attics, crawlspaces and basements, a line of eight compact, gas-fired, horizontal furnaces has been introduced by this company. Dimensions are but 19⅝ x 27⅞ x 42½ in. (44½ in. in cooling models). Compact size is cited as being made possible by a curved steel heat exchanger which offers more surface and is swept by air from two blower openings instead of one. Quiet operation is achieved by honeycombing around the single-port sheet metal burner, enabling air to mix with gas.



Four models have capacities of 80,000, 100,000, 105,000 and 125,000 Btu/hr. A second 125,000-Btu/hr model has higher air handling capacity. All can be equipped with encased horizontal cooling coils. Three other models of 80,000, 100,000 and 125,000-Btu/hr capacities are for heating only.

Carrier Air Conditioning Company Div, Carrier Corporation, Syracuse 1, N. Y.

LARGE CAPACITY UNIT COOLER

Extending the Top-Aire Unit Cooler line is a three-fan model with a capacity of 3000 Btu/hr at 10 F TD.

Other units in the series have capacities of 1000, 1300 and 1700 Btu/hr. Flat and but 4½ in. high, the unit mounts in the top of a refrigerator or in under counter cabinets, wherever space is at a premium.

Featured is an aluminum drain fitting angled at 45 deg so it can be run through the back or bottom of a cabinet. There is room for an expansion valve inside.

Bohn Aluminum & Brass Corporation, Danville Div, Danville, Ill.

ALUMINUM DIRECTIONAL DIFFUSER

Easy to install and light in weight, this all-extruded aluminum directional diffuser is available in five frame styles and fourteen core patterns. All cores are removable and interchangeable. Square and rectangular designs include one-way flow, two-way opposite flow, three-way flow and two-way corner flow. The Type D diffuser comes with snap-in, bevelled, drop collar, flange and lay-on frames.

Waterloo Register Company, Inc., P. O. Box 147, Waterloo, Iowa.

ELECTROSTATIC AIR CLEANERS

For industrial, commercial and medical applications where clean atmospheres are a critical requirement, this company has developed electrostatic air cleaners cited as removing dust particles or air pollutants with extremely high efficiencies. Air cleaners of both the plate and tube type, in capacities from 100 to 11,000 cfm, are offered. Electrical charging capacity of the units is 7 to 17 kv.

Research-Cottrell, Inc., Bound Brook, N. J.

INDICATING GAUGE

Enabling this one instrument to furnish many different readings, 84 in. of tape storage are provided in this new gauge. Designated Selecto-Scale Pneumatic Receiver, it can be used to indicate pressure, temperature, humidity and liquid levels in areas of apartments, hospitals, office buildings and commercial and industrial locations. Scale length for reading is 4½ in., but a series of scales is printed on 84 in. of laminated mylar stored on two spools. The tape can be repositioned to use other 4½-in. segments, and it can be printed to read vertically and horizontally.

Pressure elements, tape storage spools and optional Hi-Lo Alarm pressure switch assembly are assembled on an anodized aluminum drawer that slides into a cadmium-plated steel chassis. The chassis can be inserted into a cutout created for it in a panel board. Flexible tygon tubing connects the drawer assembly with the rear of the chassis, permitting partial removal of the drawer for field adjustment without breaking the air connection to the transmitting source.

American Machine & Metals, Inc., U. S. Gauge Div, Sellersville, Pa.

LIQUEFIED GASES ARE ACCELERATING GROWTH IN THE FOOD PRESERVATION FIELD PAGE 78

HOW DOES THE ENGINEERS JOINT COUNCIL AFFECT ASHRAE? PAGE 56

PRESIDENT'S PAGE

(*Continued from page 56*)

registration of all qualified engineers as distinguished from the current finder's list. The Council recommended in a report, "National Engineers Registered Study", that a massive register not be established.

9. It arranges for the exchange of official representations on EJC Board and the National Research Council – Division of Engineering and Industrial Research.

10. It sponsored the 1960 Nuclear Congress in the New York Coliseum.

Many of our members are contributing generously to EJC committee work. Our Society is well represented on all the standing committees as well as special committees. Some of the latter are International Relations; Societies' Secretaries; IAESTE (exchange of international students for technical experience); U. S. Nat. Com.—World Power Conference; Eng. Manpower Commission; Government Liaison; Eng. Information Sources; Who's Who in Eng.; Engineering Research; Technical Planning; etc.

EJC is a worthwhile activity in which professional societies participate on a national and international scale. Our Society not only receives benefits therefrom but makes contributions that add to the professional stature of our membership.

STANDARDS

Sound Standard Available

A. T. BOGGS, III

ASHRAE Associate Secretary

A proposed standard for the measurement of sound power radiated from heating, refrigerating and air-conditioning equipment has been developed by ASHRAE Standards Project Committee 36 and review copies are available on request from the Technical Secretary. Comments relative to the proposed standard should be submitted to the Technical Secretary no later than November 1.

This standard is intended to cover only the test procedure. In order that the program of sound testing may be implemented completely, interested trade associations will need to supplement the proposed standard with suitable specific directions concerning the various types of equipment to be tested, recommended rating procedures, specific ranges of ratings acceptable for each type of equipment, recommended criteria for the selection of equipment for specific types of applications, means of obtaining compliance and other pertinent items.

The proposed standard is still subject to editorial revisions as well as any major contributions of substance. Comments should include not only items having to do with the content of the standard, but also suggestions with respect to format which would help to make the presentation clearer.

BULLETINS

(*Continued from page 87*)

Capacity tables are included for the full line of twelve sizes, single-inlet, single-width design, and twelve sizes, double-inlet, double-width design. Outline-type drawings are keyed to tabulated dimensional data.
American-Standard, Industrial Div, Detroit 32, Mich.

Compressed Gas Cylinders. For oxygen, argon, carbon dioxide, refrigerants or industrial gases, these cylinders are discussed in a 26-page catalog containing specifications, sizes and recommended applications. Included are high and low pressure, seamless and two-piece cylinders. Also covered are properties of various permanent and liquefiable gases.
Press Steel Tank Company, 1445 S. 66th St., Milwaukee 14, Wisc.

Control Valves. Described in this comprehensive 64-page manual, Bulletin E600, are pneumatic diaphragm and piston actuators and a wide variety of valve body assemblies. Contained in a 20-page specifying section is information necessary for selection of a control valve. Also included are accessories, such as valve positioners, handwheels and electro-pneumatic transducers. Among topics covered are valve body and trim materials, valve plugs, pressure-temperature ratings, maximum pressure drop tables and performance data, including frequency response.
Fisher Governor Company, P. O. Box 307, Marshalltown, Iowa.

Temperature Transmitter. Discussed in twelve-page Bulletin 13-17C is the Type 12A pneumatic temperature transmitter. Features of transmitter design emphasized are: compact construction, force-balance operation and automatic compensation for ambient temperature and pressure changes. Also included are detailed specifications, descriptions of operating principle, table of temperature ranges and data on related accessories.
Foxboro Company, Foxboro, Mass.

Adjustable Speed Drives. Available in ratings from 50 through 2500 hp, these magnetic drives can be applied in many varied applications. Described in six-page bulletin 3650 are basic components of the drive, theory of operation, controls and design features.
Louis Allis Company, 427 E. Stewart St., Milwaukee 1, Wisc.

Rigid Polyvinyl Chloride. Covering rigid PVC products used in a wide variety of applications, 24-page Catalog RV-61 contains information about fabrication procedures and provides case histories of applications in various fields. Included among data given are physical properties and chemical resistance.
Kaykor Products Corporation, Yardville, N. J.

Region V to Link Technical Conference—Annual Regional Committee Meeting

To be conducted as a miniature National Meeting, Region V will hold its October 13-14 annual meeting in Columbus, Ohio.

Featured will be a Technical Conference and ASHRAE President John Everetts, Jr., as keynote speaker at the opening luncheon.

Columbus will be the host Chapter with representation by Central Indiana, Cincinnati, Cleveland, Dayton, Evansville and Toledo.

Social intervals will include events for the ladies, a dinner dance and attendance at the Ohio State-University of Illinois football game.

MINERAL WOOL INSULATING CEMENT

Effective temperature range of this insulating cement has been extended to 2100 F from a previous high limit of 1800 F. In addition to withstanding higher temperatures, the new cement is offered as being whiter than the 1800-F material which it replaces and being easier to handle. Through a reduction in module size of the mineral wool, the mixed material, ready for application, is less lumpy. Linear shrinkage after heating to 2100 F is but 3%.

Baldwin-Ehret-Hill, Inc., Trenton, N. J.

TEMPERATURE-HUMIDITY CHAMBER

Characterizing the Econ-O-Line Low-High Temperature-Humidity Chamber (Model ELHH-4-LC) is the fact that it has no mechanical refrigeration system

either for low temperature or dehumidity conditions. Utilizing liquid CO_2 refrigeration to pull temperature down as far as —100 F and to control humidity in the 20 to 95% RH range, the chamber has a high temperature range of 350 F. Internal working dimensions of the unit are 18 x 18 x 18 in. As standard equipment, it is furnished with a two-pen, two-cam programming recording controller for documentation of its performance.

Associated Testing Laboratories, Inc., Wayne, N. J.

DUCT WEIGHT RULE

Consisting of four scales printed on 0.03-in. vinyl plastics, laminated with a 0.002-in. clear plastics coating, this rule is printed with red and black lines on a white or yellow background. When the cursor is moved to a point where the hair line intersects the total footage line, the total lb and sq ft of a duct may be obtained. Duct sizes from zero to 120 in. semi-perimeter and lengths from one to 100 ft may be found.

Overall dimensions of the rule are 8 x 11 in. Included also are a galvanized-aluminum comparison chart, prominent sizes of angle iron weights, taping and insulation formulas.

Carl Raymond & Company, 3930 N. Mission Rd., Los Angeles 31, Calif.

SAFETY VALVE

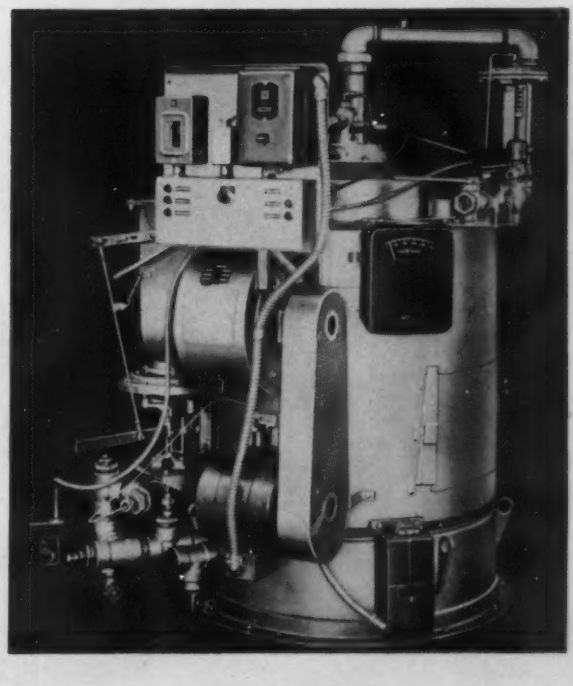
Suitable for both ammonia and halocarbon service, this new wide-range capacity safety valve is cited as having sufficient capacity to protect most types of duty encountered in refrigeration installations. It has a capacity range of from 1895 lb of air per hr at a set pressure of 100 psig through 7790 lb/hr at 425 psig.

When viewed from above, the valve body has a pear-shaped contour, which provides the gas-flow space necessary to achieve its large capacity. Gas escaping at high speed passes horizontally over the valve seat when the valve pops, sweeping the surface clean and enabling the valve to seat tightly after it has popped.

Vilter Manufacturing Corporation, Milwaukee 7, Wisc.

STEAM SUPERHEATER

Completely packaged, factory tested and adjusted, this separately fired steam superheater features auto-

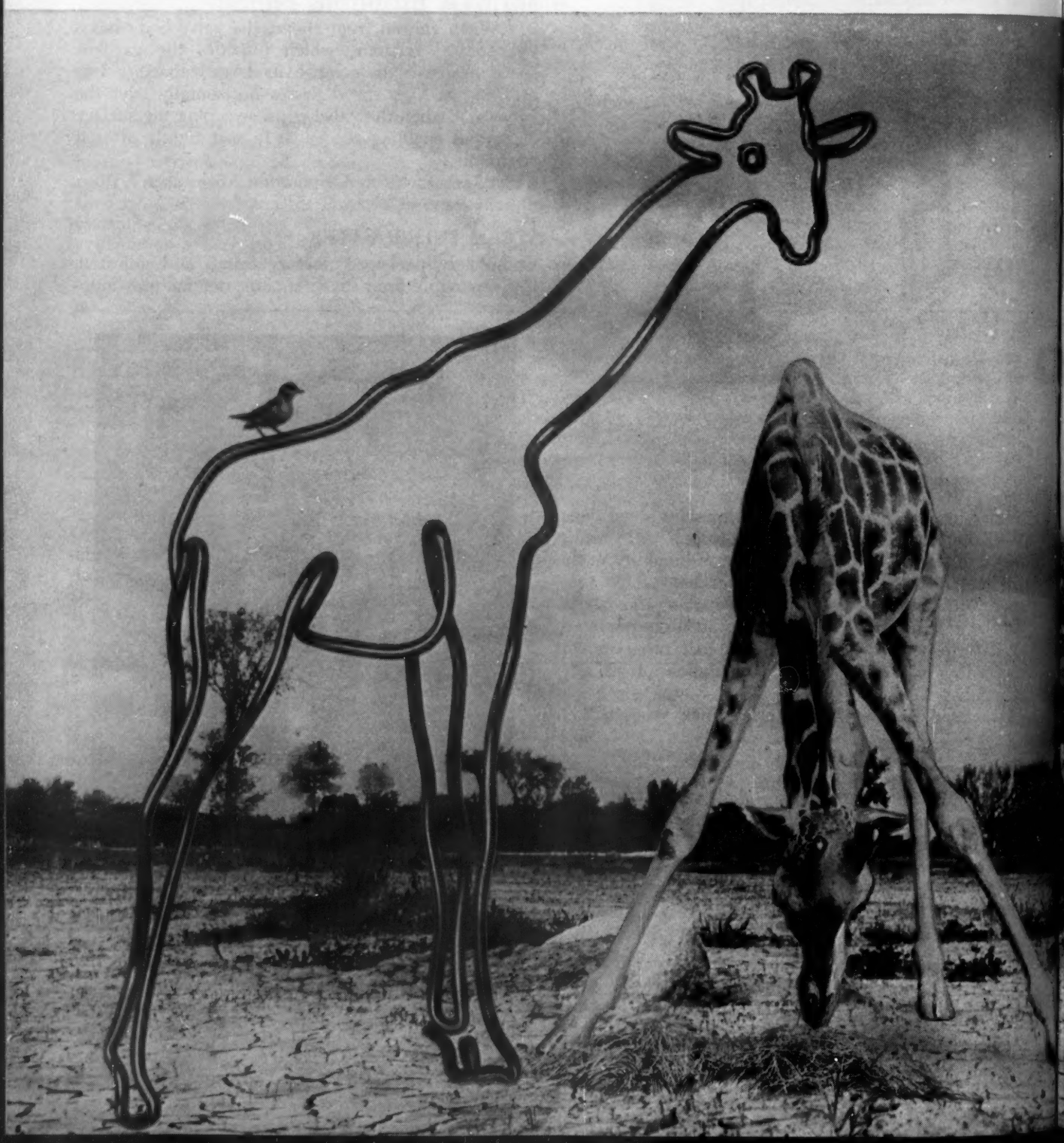


matic operation, pneumatically operated temperature control and electronic outfire control protection. Design temperatures are to 1000 F with stainless steel heat exchanger surface and to 600 F with alloy steel heat exchanger. The unit is rated at 1,000,000 Btu/hr output and can be furnished gas or oil-fired.

Vapor Heating Corporation, 80 E. Jackson Blvd., Chicago 4, Ill.

BETTER LABORATORY HOOD DESIGN WAS THE TOPIC AT THE INDUSTRIAL VENTILATION SYMPOSIUM PAGE 47

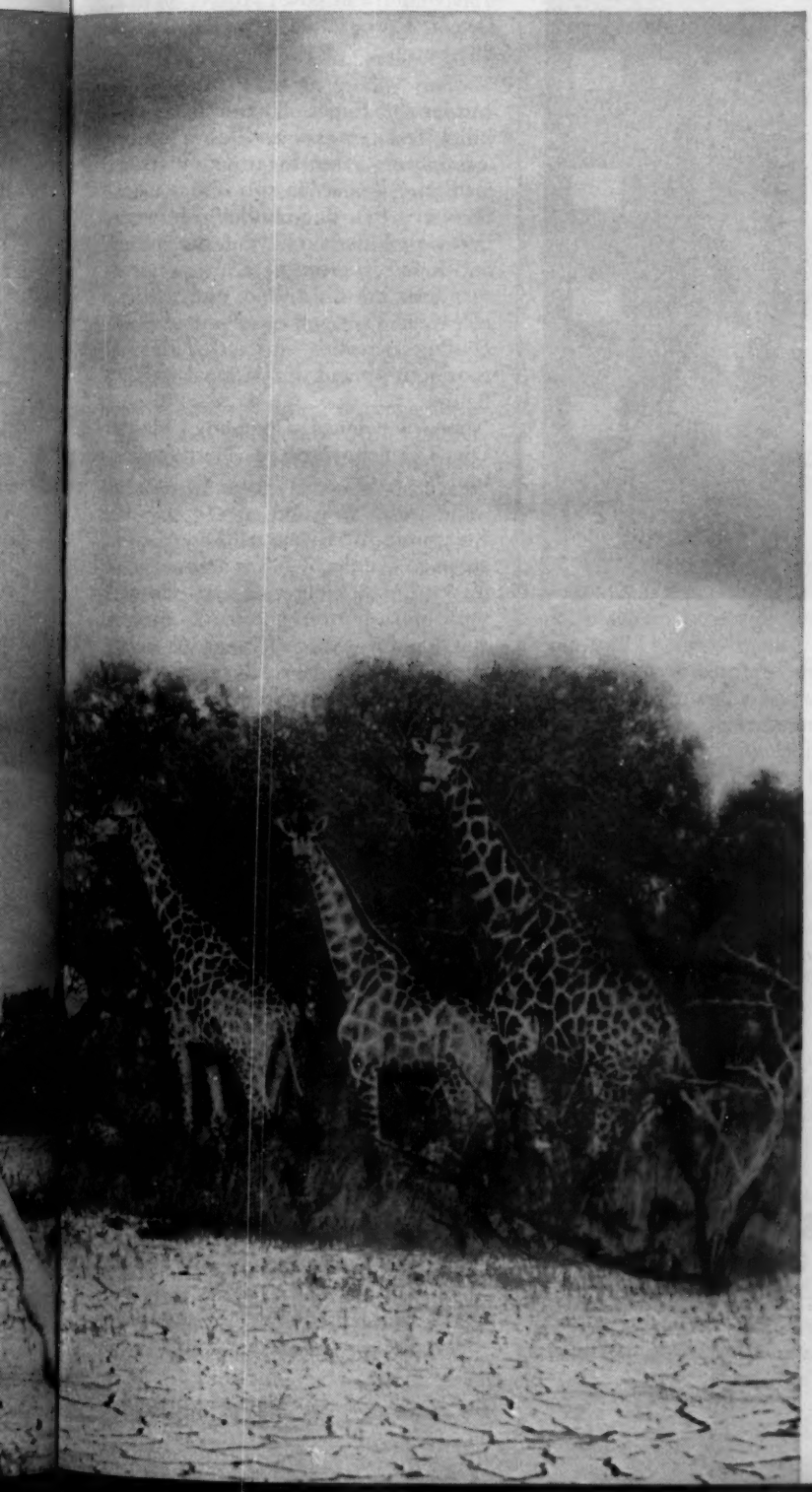
Bundy can mass-fabricate practically anything

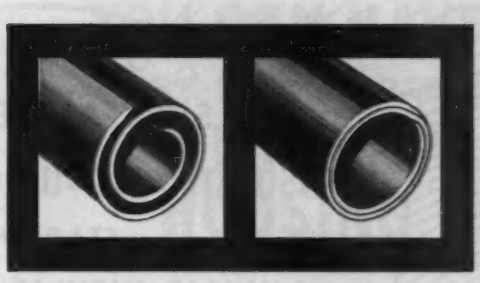


No matter how you need your tubing . . . as straight lengths, simple bends or complex coils, in orders of several hundred or several million . . . Bundy can supply it. And no wonder: There are very few things you can't do with Bundyweld® steel tubing. Standard wall thickness and O.D. are held to +.002″ to −.003″ and it meets Government Specification MIL-T-3520, Type III, and ASTM 254. With the special machines developed by our engineers, Bundy gives you every possible mass-fabrication cost advantage. Get the complete story on your tubing needs from Bundy before you buy. Call, write, or wire: Bundy Tubing Company, Detroit 14, Michigan.

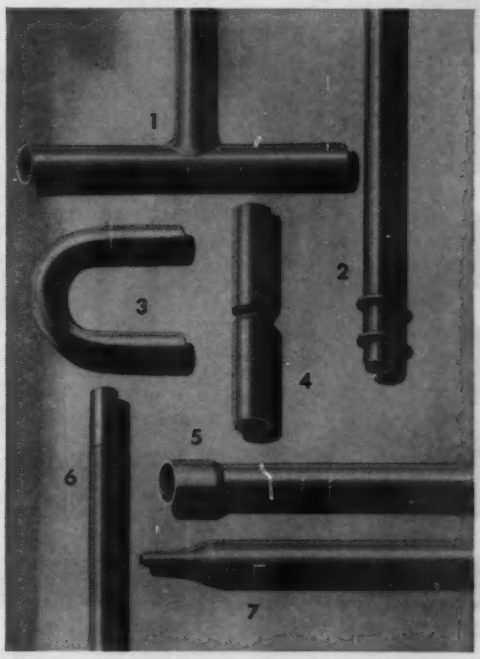
BUNDY TUBING COMPANY • DETROIT 14, MICH. • WINCHESTER, KY. • HOMETOWN, PA.

World's largest producer of refrigeration tubing. Affiliated plants in Australia, Brazil, England, France, Germany, Italy, Japan.





Bundyweld, double-walled from a single copper-plated steel strip, is metallurgically bonded through 360° of wall contact. It is lightweight and easily fabricated . . . has remarkably high bursting and fatigue strengths. Sizes available up to ⅝″ O.D.



Small-diameter tubing components mass-fabricated by Bundy may be the answer to your design problem. The Bundyweld tubing shown above is: (1) saddled and soldered, (2) double-beaded, (3) bent to minimum radius, (4) grooved, (5) expanded, (6) precision-ground, (7) swaged.

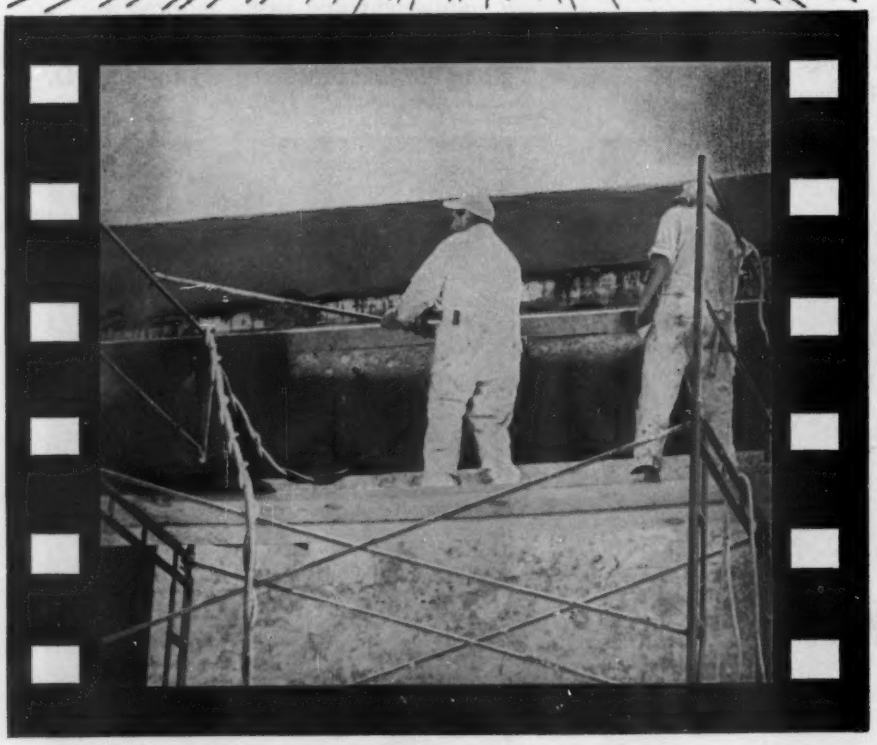
BUNDYWELD® TUBING

SPECIAL FEATURE

"INSULATED TILT-UP CONSTRUCTION"

STARRING

LAYKOLD INSULATION ADHESIVE



When builders applied the tilt-up technique to refrigerated warehouse construction, Laykold Insulation Adhesive was a "natural", adhering the vapor barrier membrane to the concrete wall panels. It also helps hold the glass-fiber blanket insulation.

Here's the way a tilt-up job goes:

1. Pour wall panel on floor and let set
2. Spray on Insulation Adhesive at 23 sq.ft./gal. It helps cure the concrete.
3. Press the vapor barrier membrane into the set Adhesive.
4. Tilt the panel up into position and anchor.
5. Spray Insulation Adhesive over membrane at 23 sq. ft./gal.
6. Press blanket-type insulation into place.

The speed and ease of this operation underscores the major advantages of Laykold Insulation Adhesive: *Spray-applied . . . cold . . . it saves time, equipment, money!*

Now Available:
a 15-minute, color-and-sound, 16mm motion picture of a tilt-up job. Ideal for employee groups or association meetings. Write for details. No charge, no obligation.

American Bitumuls & Asphalt Company

320 MARKET, SAN FRANCISCO 20, CALIF.
Perth Amboy, N. J.
Baltimore 3, Md.
Cincinnati 38, Ohio
Atlanta 8, Ga.
Mobile, Ala.
St. Louis 17, Mo.
Tucson, Ariz.
Portland 8, Ore.
Oakland 1, Calif.
Inglewood, Calif.
San Juan 23, P. R.

BITUMULS® Emulsified Asphalts • CHEVRON® Paving Asphalts • LAYKOLD® Asphalt Specialties • PETROLASTIC® Industrial Asphalts

BULLETINS

Space Heaters. Industrial forced air space heaters, designed for use in buildings where frequent changes of air are necessary, have been introduced by this manufacturer in six-page Bulletin 580-14. Designated Directflow, the gas-fired units are suited for applications where noxious gases, acid fumes, dust or hot vapors are released as part of the manufacturing processes and expelled by exhaust fans. Seven models are available, ranging in air output capacity from 11,000 to 60,000 cfm and in heat output to 6,750,000 Btu/hr, with a maximum turndown ratio of 25 to 1. **Dravo Corporation, Neville Island, Pittsburgh 25, Pa.**

Insulating Foam. Dimensional stability, reduced cost and lower density are features cited for a new urethane insulating foam for sub-zero applications in a four-page bulletin. Information is provided on advantages, mixing and K-factors, results of low temperature tests are illustrated, and a listing of physical properties is provided, including densities, strengths, thermal conductivity and dimensional stability figures.
Nopco Chemical Company, Plastics Div, 175 Schuyler Ave., North Arlington, N. J.

Electronic Air Cleaners. Discussed in six-page Catalog 1435 is a new line of heavy-duty, electronic air cleaners with push-button automatic control, suitable for a broad range of industrial and commercial applications. Sections of the booklet cover principles of electronic air cleaning, construction features, selection data and details on specific application arrangements.
Westinghouse Electric Corporation, P. O. Box 2278, Pittsburgh 30, Pa.

Wiring Block. Detailing specifications on a new modular terminal panel wiring block, four-page Bulletin 465 describes such advantages as extremely high density per lineal in. of track, modular design to permit versatility in circuitry stack configurations and tool-free insertion, withdrawal and changing of circuit leads. Dimensions are provided for the various component parts (track, cage assemblies, terminals, insulation supports and end barriers) of both Series 3 and 4 Termi-Blok connector sizes. Extensive test data also are given.
AMP Inc., Eisenhower Blvd., Harrisburg, Pa.

ASHAE HOMER ADDAMS AWARD for 1960 was to Alan B. Wagner, graduate student, Case Institute of Technology and consisted of $600 cash and a certificate. Reference to this annual award to a graduate student on an ASHRAE Research Project was inadvertently omitted from the news coverage of our 68th Annual Meeting in the preceding issue of the JOURNAL.

Heat Exchangers. Having an externally packed floating head, removable tube bundle heat exchangers are the subject of six-page Bulletin O-1162. Cut-away photographs and cross-section drawings show important construction features, including double O-ring seal protection, which is standard on Type OP units with shell diam from 6 through 16 in. Sizes, dimensions and other specifications for Type OP exchangers, as well as for larger Type SP models with shell diam from 18 through 32 in., are covered extensively.
Basco, Inc., North Tonawanda, N. Y.

Acoustical Turning Vane. To reduce noise in air distribution systems, Sono-Turn vanes have been introduced, consisting of a perforated aluminum surface with a core of sound-absorbent material. Descriptive of the unit, four-page Bulletin ST-61B contains performance graphs, an estimating chart and information on product features.
Sound Control Products Company, P. O. Box 11179, Station A, Palo Alto, California.

Duct Heater Assemblies. New gas-fired duct heater assemblies for complete or supplemental heating of industrial and commercial buildings are discussed in four-page Bulletin GA-61. Advantages and specifications are given, units are illustrated and tabulated data provide outlines, dimensions, capacities, mounting height and floor area coverage.
L. J. Wing Manufacturing Company, Linden, N. J.

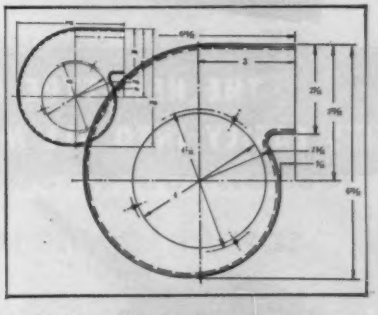
General Purpose Motors. Designed for such applications as driving compressors, pumps, high inertia fans and blowers, refrigerators and air conditioners, a line of ac, general purpose, integral hp motors is discussed in six-page Bulletin L-3314A. Single-phase motors range to five hp, open drip-proof, and to two hp, TEFC; three-phase ratings are to ten hp open and three hp TEFC. Single-phase motors are capacitor start and three-phase are squirrel cage designs for high starting torque, low starting current, constant speed operation. Contained in the bulletin are selection and specification tables, mounting-arrangement drawings, speed-torque curves, principal dimensions and special designs.
Howell Electric Motors Company, 16316 W. Seven Mile Rd., Detroit 35, Michigan.

Make-Up Air Systems. Depicting six basic types of Sun-Flo direct gas-fired make-up air units, in a total of more than 40 models with capacities from 5000 cfm with 750,000 Btu/hr to 150,000 cfm with 15,000,000 Btu/hr, is 16-page Bulletin C-101. Units are available for roof-top, ceiling or wall mounting. Also shown are nine models of burner sections and make-up air system accessories, including shut-down damper sections, door heat seals, angle filter banks, roof hood air intakes, wall intake louvers and louver towers.
Metals Engineering and Manufacturing Company, Inc., 8824 Lyndon, Detroit 38, Mich.

Refrigeration Valves. Describing 292 different valves designed for ammonia refrigeration, L-P gas service and general industrial applications is twelve-page Catalog 203. Emphasis has been placed on the company's line of ductile iron valves. Included are globe and angle shut-off valves as large as six in. with interchangeable threaded, socket-weld and welding neck flanges; pressure relief, flow check and hand expansion valves; strainers, liquid level gauges, charging hose sets, quick disconnect couplers and a line of forged steel and flanged unions.
Henry Valve Company, 3215 W. North Ave., Melrose Park, Ill.

Single-Phase Motors. Bulletin 300 covers a line of integral (one through five) hp, capacity start, induction run, single-phase open motors. Starting torques are from 250 to 350% and overload capacities average 250% of full load torque. Units are suited for such general purpose applications as driving air conditioning units, blowers, air compressors and coolers. Illustrated in the bulletin are a variety of design features. Typical performance curves are included and Nema frame assignments are listed by hp and speeds.
Leland Ohio Electric Company, a Subsidiary of Howell Electric Motors Company, 16316 W. Seven Mile Rd., Detroit 35, Mich.

YOU CAN HAVE
PRE-ENGINEERED
BLOWER HOUSING ASSEMBLIES



BUILT-IN DE-STA-CO QUALITY

QUICK DELIVERY

Long experience and tooling flexibility provide quick delivery of pre-engineered Blower Housing assemblies. Broad range of sizes and styles—in any width—available from stock dies in a matter of days . . . and no tooling costs! For special installations ask our engineers how adaptations can be made at low unit cost.

WRITE FOR FULL DETAILS AND SPECIAL BROCHURE

DETROIT STAMPING COMPANY
340 MIDLAND AVENUE · DETROIT 3, MICHIGAN

TOGGLE CLAMPS · STAMPINGS
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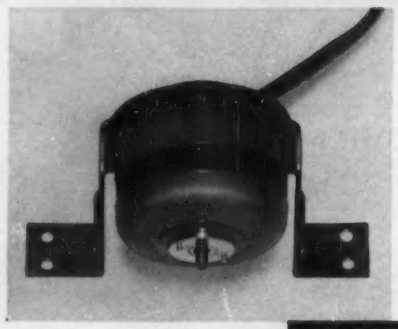
SHIMS AND SPACERS · BLOWER HOUSINGS

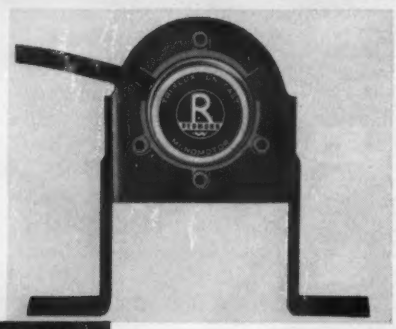
Completely Interchangeable

NEW

Accommodates

THE NEW TYPE AM MONOMOTOR®
FITS EASILY INTO ALL MOUNTINGS AND BRACKETS

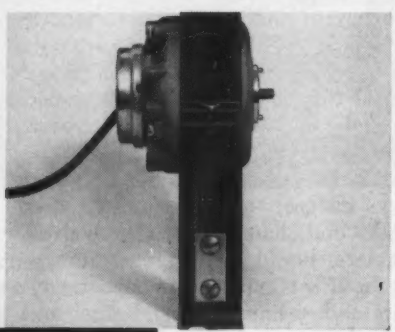




REAR MOUNTING

3/8ths inch smaller diameter allows motor to fit right into bracket now in unit—no bending, squeezing, pinching.

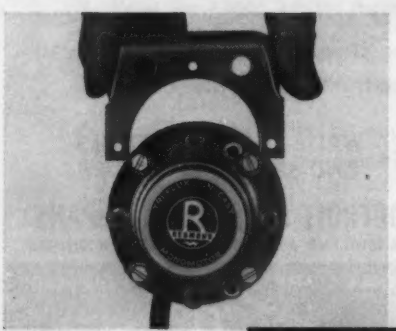
Fits all rear-mounting applications. 6 mounting holes make it easy to mount motor on the bracket.



PERIPHERAL MOUNTING

You can easily replace motors with peripheral mounts or that are lug-mounted.

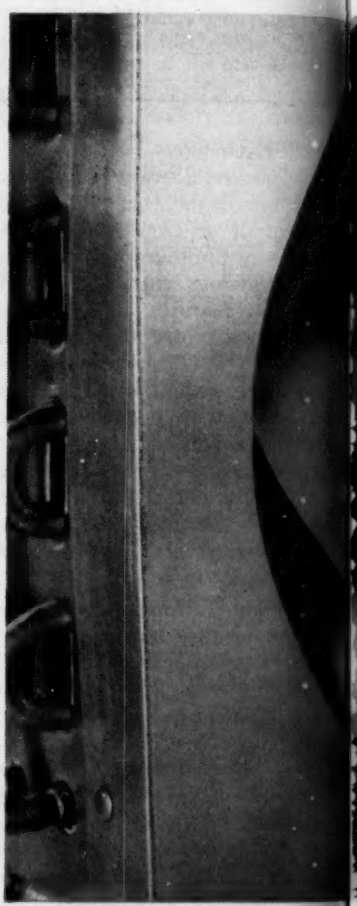
In a peripheral mount, the AM motor can be adjusted to any depth.



PEDESTAL MOUNTINGS

Pedestal bracket fits right over motor—note number of holes for affixing.

Pedestal bracket attaches easily in any of numerous combinations available.

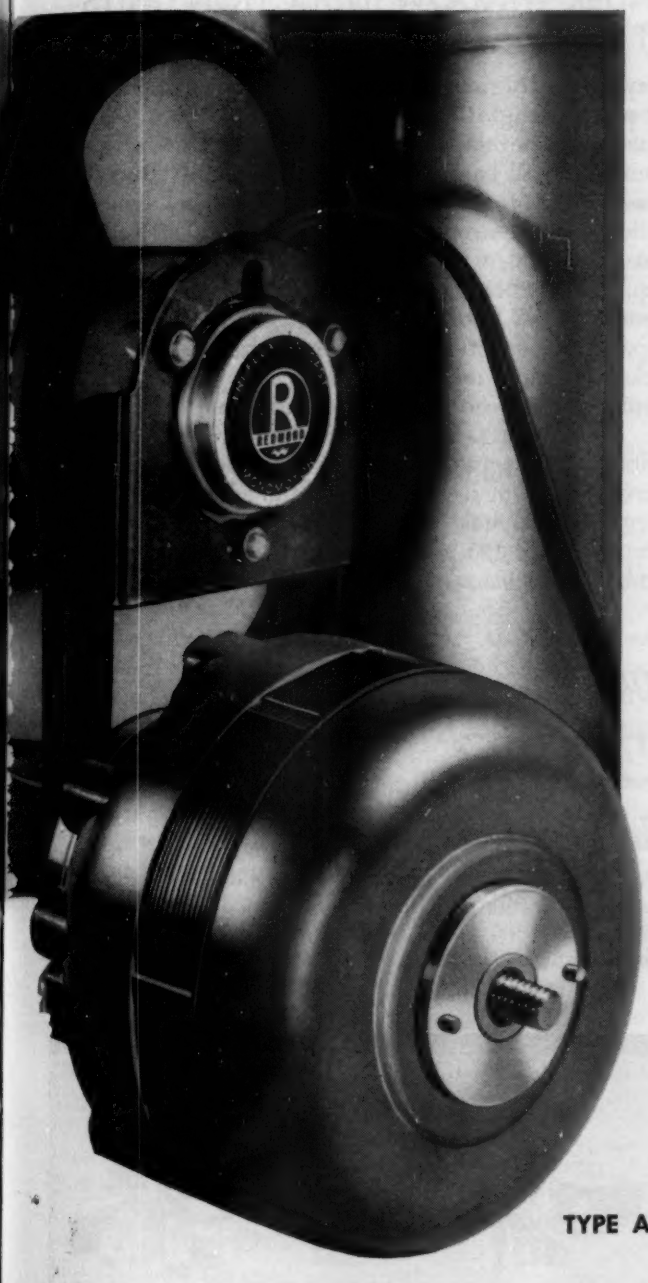


FREE
BROCHURE

Type AM Catalog Page has complete information: dimension diagrams; model numbers; sizes; weights. Send request to Distributors Division, Redmond Company, Inc., Owosso, Michigan.

DISTRIBUTORS DIVISION

Redmond Condenser Fan Motor All Mountings



TYPE AM

These Features Make The New Single Bearing Type AM Your Number 1 Condenser Fan Replacement Motor

As you can see from the photographs at left, now you need to stock only this one motor to accommodate all mountings and brackets—rear, peripheral, and pedestal.

The diameter of the new Type AM MonoMotor® has been reduced 1/8ths of an inch, so it fits right into rear-mounting brackets without any bending or knuckle-busting. The smaller size and lighter weight make it easy to handle when you are working on a unit. And of course the fact that it is up to 25% lighter in weight than are competitive motors means real savings in shipping costs for you.

When you install motors that last, you get satisfied customers. Life tests prove that the Type AM has longer life than competitive motors. One of the reasons that the AM gives longer, trouble-free operation is the exclusive Redmond patented oil system, which provides more effective oil recirculation. The new AM has the best oil slingers in the industry. Redmond's patented features guarantee no oil leakage in use or in shipment.

Your customers will like the new AM MonoMotor® because it is whisper-quiet. Extremely close manufacturing tolerances combined with Redmond's patented controlled rotor end play feature eliminate the end play noise problems found in conventional motors.

All-angle operation is another feature of the AM—it is designed for all-position mounting, vertical shaft up, shaft down, or any angle.

The Standard of Dependability

Redmond Company, Inc. OWOSSO, MICHIGAN

Subsidiary of

COOKSVILLE, ONTARIO • **CONTROLS COMPANY OF AMERICA** • ZUG, SWITZERLAND

Applications

AMMONIA EVAPORATORS FREEZE BAKERY PRODUCTS

Kitchens of Sara Lee, Inc., producers of frozen bakery products, utilize Kramer Trenton evaporative refrigeration equipment in their new Chicago bakery. Products coming from the ovens are first cooled to room temperature, then transported to a blast freezer, where they are frozen in the package, and finally moved into a 144,000-cu ft, –5 F storage room to await distribution via refrigerated truck. Selected for the blast freezer were four ammonia evaporators with automatic re-evaporative defrost. Maintaining the storage room at constant temperature are two ammonia evaporators with re-evaporative defrost and a duct distribution system.

High side equipment consists of two 75-hp compressors and two 50-hp rotary boosters. Refrigerant is sub-cooled in a constant level accumulator and circulated by the booster pumps to the low temperature evaporators. All evaporators are controlled by room thermostats with back pressure regulators in the suction lines of the room.

Low temperature portion of the system is of the recirculating type, operating at an 8 F temperature difference, while the medium temperature part is on direct expansion at 10 F TD. An electronic instrument is installed in the machine room to indicate air temperatures throughout the building. Air at 50 F is circulated under the floors through tile ducts. During the summer it is discharged internally and in winter to the outside.

UNDER WINDOW INDUCTION UNITS AT UNITED ENGINEERING CENTER

Varying heating and cooling output by changing the air path through the unit, new under-window induction units, developed by Carrier Air Conditioning Company, will be utilized in the 20-story United Engineering Center. A sensitive damper in each controls airflow from the room through or around the heating or cooling coil, with a change in heating or cooling requirements being met by sending a larger or smaller amount of air through the coil. In addition to the bypass air control, the discharge grille sections are locked in place and a redesigned cooling coil arrangement permits easy cleaning from the front of the unit.

Chilled water for the 375 Bypass Weathermaster units and the central apparatus will be supplied by a Carrier centrifugal water chiller and an absorption refrigerating machine working in tandem to deliver 900 ton. Steam at 125 psi will enter the turbine to drive the centrifugal machine.

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CIRCULAR SHOWROOM SERVED BY UNDERGROUND DUCTS

Circular in shape, with floor-to-ceiling glass and an eight-point, star-shaped roof (causing a variance in floor-to-ceiling elevation from 14 to 20 ft), the new Rayco retail store in Paramus, N. J., presented a problem in location of ductwork. Overhead ductwork was not feasible because of the finished interior of the roof and exposed beams.

Two 14-in. ducts were laid in concrete trenches under the showroom floor, extending to form a large V. From these, eight smaller ducts fan out to the perimeter of the structure, where air is diffused through grilles in the floor, blanketing the glass walls with conditioned air. A few side-wall registers were fitted at the rear of the showroom. Air enters the ductwork at the apex of the V, where a 15-ton direct expansion fan-coil unit is suspended from ceiling beams. Directly above this unit, on the roof, is a low-silhouette, 15-ton Carrier condensing unit. Also attached to the system is a gas-fired, 300,000-Btu/hr duct furnace, adding winter heating.

662-TON COMPRESSOR SYSTEM FOR 1,000,000-CU FT BUILDING

Headquarters of the Maytag Company, Newton, Iowa, has 1,000,000 cu ft of air conditioned space, served by two 331-ton Carrier compressors. Automatic transistorized controls regulate the building's 261 separate temperature zones. A master control panel, located in the penthouse, automatically regulates all heating, cooling, dehumidifying, humidifying and mixing functions; this panel can read temperatures in 53 zones of the building, 33 of which can be adjusted manually by wall-mounted thermostats. All air is filtered by an American Air Filter Company electrostatic precipitator, which will remove particles of dust, smoke and pollen as small as ¼ micron.

Temperature and humidity in the building are maintained at 76 F and 35 to 50%, respectively, winter and summer. Air circulated totals 193,892 cfm, with a complete circulation of air in any area every six min. Every 30 min all air in the building is removed and replaced with fresh outside air, a total of 36,515 cfm of air being introduced. Separate exhaust systems operate in restrooms and laboratories.

RADIANT HEATING WARMS SPECTATORS IN SPORTS CENTER

Sixty rows for spectators in the 5000-seat arena of a new sports center in Montreal, as well as sections alongside the 85 x 190-ft ice rink, contain radiant heating. One-in. diam A. M. Byers wrought iron pipe specified for the system was fabricated into sinuous coils, laid on six-in. centers in the spectators' section and then encased in two in. of concrete. Coils alongside the rink were spaced on six-in. centers and covered with two in. of concrete with terrazzo surface. Water will be heated to 110 F and circulated through the piping.

Two heating zones, each having its own converter and control, comprise the heating system. The heat exchanger will provide 2,500,000 Btu/hr, using 2700 lb of steam per hr to heat water. Each circulator will provide 125 gpm against 20-ft head.

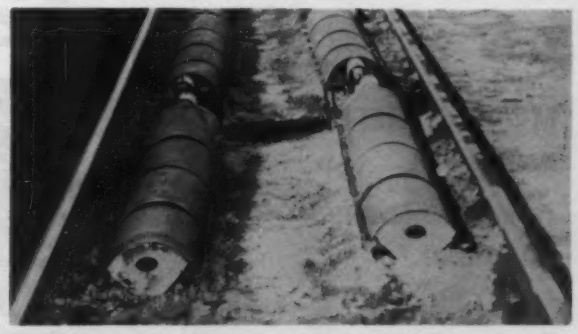
For the ice rink, wrought iron piping also was specified. 55 ton of 1¼-in. diam 4-D wrought iron pipe is spaced on 4½-in. centers. Supply and return piping is eight in. diam. Operating the rink will be 150 ton of refrigeration. A brine solution will be circulated through the piping, pumped from brine tanks located beneath the rink.

INDOOR SKATING RINK USES 29,000 FT OF PLASTICS PIPE

Conveying brine at a temperature of 22 F, a network of 29,000 ft of plastics pipe is contained within an indoor ice skating rink measuring 148 x 57 ft. Used is Republic Steel Corporation's one-in. FE, a flexible polyethylene pipe rated for 75-psi working pressure at 75 F. Feeder lines first were placed along the sides of the rink. Plastics pipe then was cut into lengths of 56 ft, set in parallel rows across the width of the rink and connected with the feeder lines. Insert fittings were used to join the pipe to the feed lines. A stainless steel clamp was slipped over each of the pipe ends and a coupling was then forced into the two ends. Worm-drive screws were tightened with sufficient pressure to cause the serrated ends of the coupling to impinge against the ID of the pipe.

UNDERCAR HEATERS REDUCE THAWING TIME

Installation of a new type of undercar heater at Colorado Fuel & Iron Corporation's Pueblo Plant is cited as reducing thawing time of hopper cars con-



taining frozen coal by as much as 75%. During the winter, temperatures as low as –20 F require thawing of coal prior to its being supplied to the plant. Formerly thawing was done with gas torches placed under the cars, a method requiring up to four hr to thaw two carloads. To speed up thawing, the company installed 16 Radiarc Heaters, produced by Hauck Manufacturing Company, rated at 500,000 Btu/hr. These gas-fired units provide radiant and convection heat simultaneously.

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SALES ENGINEER — Challenging opportunity for growth with leading manufacturer of heat exchangers, pumps, and hot water heating equipment for man experienced in selling specification equipment to wholesalers, contractors, and engineers. Engineering or technical degree or equivalent experience. Willing to relocate and travel. Send resume to Box 202, ASHRAE, JOURNAL.

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